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Assessment of the Technical and Economic Feasibility of Metal Matrix Composite Heavy-Duty (Class 7/8) Brakes

Prepared for:

Transportation Development Centre
Transport Canada
Montreal, QC

Centre for Surface Transportation Technology
National Research Council Canada
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16. Abstract

This study provides a broad technical review and competitive analysis of lightweight metal matrix composites (MMC) of aluminum in heavy-duty vehicle brake drums and rotors. It reviews current brake technology and markets, and provides information on existing brake characteristics, such as wear and cost. Conventional cast iron materials are compared to homogeneous and composite mixtures of aluminum and silicon carbide or alumina. The results of thermal and stress modelling analyses are presented, estimating the performance of the different materials under realistic extreme braking conditions. A manufacturing cost model is developed to analyse comparative manufacturing costs, and to investigate how these cost structures are affected by cost inputs and production volumes.

Brake manufacturers are not currently pursuing development of aluminum MMC brake products, largely because of past technical failures and the perceived high cost of the materials. This study concludes, however, that the use of aluminum MMC materials for heavy-duty vehicle brake rotors is technically and financially feasible, thanks to new material formulations and production techniques.

It is recommended that additional physical testing of the new material formulations be conducted to validate these conclusions. To this end, cost estimates are provided for a prototype testing program, along with a list of potential industrial partners interested in the continued technical development of such products.

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16 **Résumé**

Ce rapport rend compte d'une vaste étude technique et d'une analyse du marché qui ont porté sur l'utilisation de composites à matrice métallique (CMM) légers à base d'aluminium dans les tambours et disques de freins pour véhicules lourds. Il examine la situation actuelle de la technologie et du marché des freins, et donne une idée de certaines caractéristiques des freins de type courant, comme leur taux d'usure et leur coût. Il compare les matériaux classiques, soit la fonte, aux mélanges homogènes et composites d'aluminium et de particules de carbure de silicium ou d'alumine. Il présente ensuite les résultats d'analyses théoriques de la capacité thermique et de la résistance aux contraintes des matériaux, dans des conditions réalistes de freinage extrême. Enfin, un modèle de coût de fabrication permet de comparer les coûts respectifs de diverses options de fabrication et de mesurer l'effet du coût des intrants et des volumes de production sur le coût de fabrication.

Si les fabricants de freins écartent pour l'instant l'option de freins en CMM à matrice d'aluminium, c'est beaucoup en raison des échecs techniques qu'ont essuyés ces freins par le passé et du coût élevé qui semble rattaché à cette option. Pourtant, la présente étude conclut que l'utilisation de matériaux CMM pour les disques/tambours de freins pour véhicules lourds est techniquement et économiquement viable, grâce à des formulations de matériaux et à des procédés de fabrication améliorés.

Les chercheurs recommandent de mener d'autres essais portant sur les nouvelles formulations de matériaux pour valider les conclusions ci-dessus. À cette fin, ils ont établi les coûts estimatifs d'un programme d'essais sur prototypes, de même qu'une liste de partenaires du secteur privé qui pourraient être intéressés à participer au développement de tels produits.

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EXECUTIVE SUMMARY

This study was commissioned by the Transportation Development Centre of Transport Canada and is aimed at providing a broad technical review and competitive analysis of the use of lightweight metal matrix composite (MMC) materials in heavy-duty vehicle brake drums and rotors. As brake systems are typically a high maintenance-cost item to operators, especially for urban transit operators, the potential benefits of advanced materials in reducing wear, weight and maintenance incidence could be highly advantageous in decreasing costs, improving safety and decreasing energy consumption.

The materials that are specifically of interest in this study are MMCs of aluminum. There are two forms of this matrix: one that has been developed as a premixed, homogeneous composite of aluminum and silicon carbide (SiC/Al) by Alcan Ltd. under the trade name of Duralcan;[1] and a second non-homogeneous composite in which preformed inserts of a secondary material, usually alumina (Al_2O_3), are infiltrated with aluminum. The Materials Technology Laboratory of Natural Resources Canada (NRCan) has some proprietary technology in this material. In both systems, the goal is to introduce a strengthening and heat transfer medium into the aluminum so that it can be used in a greater range of stress and heat applications.

The use of lightweight materials in vehicles has been increasing as the needs for high fuel efficiencies and higher vehicle performance increase, and as the fabrication and application costs of alternate materials decrease. The amount of lightweight materials used in transport vehicles is directly proportional to the value the manufacturer or end user places on weight. Therefore, airplanes are largely made of lightweight materials such as aluminum, titanium, plastics, etc. Road vehicles have a lower content of more expensive lightweight materials, again depending on their application and market.

Traditionally, brake drums and rotors for both light and heavy vehicles have been made with cast iron. The primary advantages of cast iron are its wide availability and low cost. Disadvantages include its weight, which causes a reduction in vehicle fuel efficiency; its physical characteristics, which increase vehicle noise, vibration and harshness; and its relatively high wear rate, relative to some of the possible alternatives, which causes a high frequency of adjustment for wear.

With the development of new composite materials with improved properties, composite brake rotors (which overcome most of the drawbacks in cast iron brake rotors) are now manufactured and employed in light vehicles. In this study, the viability of extending the utilization of composite brake rotors to heavy vehicles is evaluated. Specifically, the study focusses on whether composite brake drums and rotors can meet the service, manufacturing and market entry challenges.

The study was divided into four major parts:

1. A review of brake technology and current markets for heavy vehicle brake components and an estimation of the possible economic value of the new brake materials attributes (weight and wear).
2. An analysis of the alternative materials and their physical capabilities to be used for heavy-duty truck brake drums and rotors.
3. An economic modelling of production costs for the alternative materials.

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[1] Other aluminum producers, such as Norse Hydro, have formulated similar MMC materials.

4. An analysis of the technical and economic probability of a successful brake drum or rotor product and a development plan to achieve success.

The findings in each of these work areas are summarized in the following text.

Market Analysis

Demands on braking systems have been increasing dramatically over the past 50 years. It is estimated that there has been a 30 percent increase in the braking power needed as a result of:

- higher posted speeds,
- higher gross vehicle weight (GVW) limits,
- a 30 percent decrease in the vehicle's natural retardation versus 15 years ago,
- a 19 percent reduction in engine braking force, and
- a trend to low-profile vehicles.

The tremendous braking forces needed to stop a heavy truck generate a correspondingly high amount of heat. Truck brakes are obviously much larger in absolute size than passenger vehicle brakes, but in relation to vehicle weight they are smaller. In particular, the contact surface of the brake drum (the "swept area") is considerably smaller (about a half to a quarter of the size relative to vehicle weight) in air-braked combination units than in passenger cars, as seen in Table 1. Since truck brakes are relatively smaller (in relation to vehicle weight), they have correspondingly less thermal capacity than do passenger vehicle brakes. Also, as the application forces are much higher, the specific power that they have to absorb can be 400 percent greater than a car.

Table 1 Comparison of Passenger Car and Truck Disc Brakes

Vehicle type	***Brake lining area cm^2/kg of GVW***	***Swept area cm^2/kg of GVW***	***Specific power kW/m^2***
Passenger Cars	0.40	1.21	67
Tractor Trailers	0.36	0.57	300
Ratio	0.9	0.5	4.5

There are two basic types of brake rotors: drum and disc. Drum brakes were used on the front and rear wheels of most cars and trucks prior to the 1970s. They are still used extensively for the rear wheels of production cars and on all commercial vehicles outside of Europe. Disc brakes have greatly increased their market penetration in passenger cars and in Europe are used for most new trucks and trailers.

The total value of the brake drum and rotor market in North America and Europe is estimated at close to US$1 billion in each. The market for brake components is supplied through parts made or sold to the Original Equipment Manufacturers (OEMs) or through the aftermarket supply channels. In North America all OEM parts are supplied by third party suppliers. In Europe, OEMs manufacture about 50 percent of their components, but this percentage is shrinking quickly with Tier 1 supplier shares gaining. However, the brake suppliers are highly in a few companies. The largest supplier is ArvinMeritor, which has about 80 percent of the new product market in North America. In the aftermarket, there are more suppliers participating, but the bulk of the brake drum market (over 80 percent) is supplied by three or four suppliers (ArvinMeritor, Webb Wheels, Hayes Lemmerz, Gunite). Because of the large size and competitiveness of the brake drum market, virtually all Tier 1 suppliers purchase the rough cast drums from casters. They then test, machine, package and distribute the product.

The North American and European marketplaces are significantly different in a number of ways:

- The buyers specify the brake system components (or supplier) in North America when ordering a truck or trailer, whereas in the European Union (EU) the OEM has total control of the brake system design.
- EU brake standards are different, having essentially forced a shift to disc brakes because of higher energy loading demands, faster actuation times, etc.
- There are more small fleets in North America compared to the EU and thus less central fleet buying. There are also fewer engineering resources for product specification or testing.

The difference in regulated braking requirements between US (and Canadian) and EU new product regulations has played a significant role in shifting the EU market to discs and electronic braking systems. EU performance regulations emphasize high-energy braking, braking stability and fade more than the US regulations and these demands are better met by disc brakes, which have inherently less fade as well as fast application and release times. This pressure, plus different product bases between drum and disc suppliers in Europe, has caused the new truck brake market to shift within the past five years to approximately 80 percent discs on tractors and straight trucks and 50 percent on trailers. In North America, discs account for less than 1 percent of the market. It is expected that disc brakes will attain a market breakthrough in North America only if regulations are changed to require their superior performance capabilities.

Once the new brake material's technical capability is proven, the market will accept the application of lightweight brake technology on its direct economic value to the operator. These benefits lie primarily in the value of weight reduction. Current evidence from the marketplace suggests that about 20 percent of the market is very weight sensitive and will pay up to C$10/kg for weight reduction. Some smaller niches will pay even more. If the differential costs can be reduced to C$3/kg, then significant penetration of the mass truck market could be expected.

The fact that MMCs and carbon-carbon composites have the potential to greatly reduce the wear rates is thus a very important physical attribute of the materials. The magnitude of the impact of changing the wear rate was estimated for a single or dual drive axle truck (or bus) and indicated that reducing wear rate by 25 or 50 percent (increasing time to replacement by 50 and 100 percent) could reduce the likelihood of brakes being out of adjustment by 80 and 83 percent. In comparison, automatic slack adjusters (which are now mandated as standard equipment) reduce the out-of-adjustment probability by 75 percent. This change in wear rate is well within the potential effect of the material change (2 to 10 times decrease in wear rate) and provides strong evidence that the brake performance, on average, could be greatly improved by the use of low wear rate materials.

The impact of potential weight saving on fuel consumption is a function of the duty cycle on which the vehicle is used and its configuration. Empirical and engineering model evidence suggests that line-haul operations would have the smallest impact at 1.2 percent change in consumption per 1000 kg reduction. Straight trucks in urban delivery operations and transit buses are expected to achieve 2.4 and 3.4 percent consumption reductions per 1000 kg weight reduction, respectively.

Material Physical Analysis

The material physical analysis was undertaken on a number of levels:

1. Review previous literature on brake testing and modelling, performance analysis of MMC materials.

2. Review basic material properties corresponding to the functional performance of a brake rotor.

3. Review current composite brake rotor manufacturing methods for light vehicles.

4. Construct computer models to predict both stress and temperature profiles experienced by brake rotors in heavy vehicles using the candidate materials.

From the material standpoint, composite materials offer superior physical, thermal and mechanical properties, as well as lighter weight. The use of SiC/Al in brake rotors was limited by its low maximum operating temperature (MOT) of 450°C. CANMET-modified Al_2O_3/Al composite promised to have a much higher MOT of around 600°C, but full dynamometer test data are not yet available. Carbon/silicon carbide (C/SiC) showed the best performance, having a MOT of 1350°C.

Manufacturing analysis indicated that all materials could be manufactured in the sizes and shapes required. Both the aluminum materials were capable of being cast using squeeze casting techniques instead of high-pressure casting. This reduces the costs of the casting process for each. The C/SiC material has been manufactured in rotor configurations in relatively small production volumes.

Results of the computer modelling (with extremely conservative conditions) work at CANMET suggests that the temperature of aluminum composite brake rotors in heavy vehicles can reach as high as 615°C for disc brake rotors (thermal history example shown in Figure 1), and 447°C for drum brake rotors. Based on this conservative estimate, the use of SiC/Al for brake rotors in heavy vehicles is suitable for drum brake rotors but not disc brake rotors. As well, CANMET Al_2O_3/Al composites have the potential of meeting the stringent service environment for the drum brake application, but not necessarily the disc brake application. However, the CANMET material analysis is less certain because material property data confirmation tests – required for a more certain analysis – on the Al_2O_3/Al composites were not available.

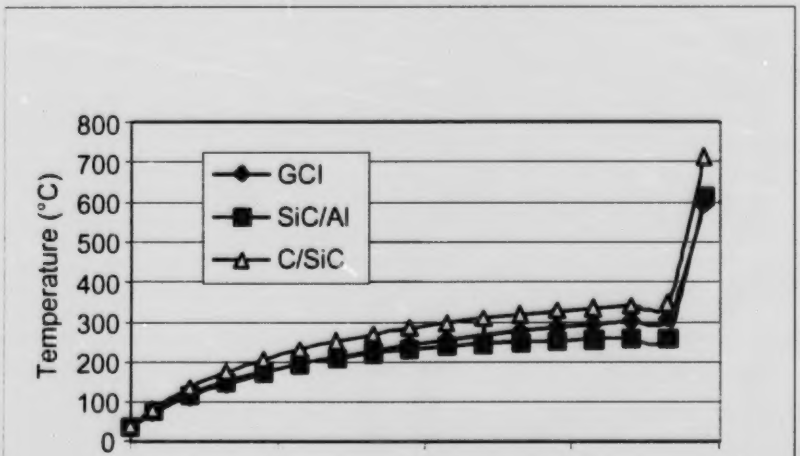

Figure 1 Computer-Modelled Disc Rotor Temperature History

With a MOT of 1350°C, C/SiC is definitely the most physically attractive composite of the three materials for use in heavy vehicles. In addition, results of the stress analysis indicate that all three composites can sustain the stresses developed in the rotors during severe braking conditions.

During the course of this study, Volvo Technology Development Corp. published information on improving thermal performance of MMC materials through the use of slightly wider particle sizes for the 20 percent SiC in the aluminum and through chemical etching of the wear surface of the finished casting. The brake chosen for its development work was the front brake for a Volvo S40 with 15-inch wheels. The size was constrained to allow direct substitution with the existing cast iron disc rotors. The work is judged to be very significant to the future use of MMC in brakes.

A prerequisite for proper function of an MMC-based brake is a stable friction transfer layer (the layer between the drum or disc and the brake pad). Volvo (and others) have found that different pad formulations have different abilities to create this transfer layer. Prior research has shown that the stability of the transfer layer is not only determined by the pad material but is also strongly dependent on the structure of the disc surface.

In the Volvo program, tests with the same pad formulation showed that some of the discs could withstand temperatures above 450°C without any problems while other discs, also with a fairly good particle distribution, suffered from surface scoring at temperatures below 400°C. Investigations of the disc surfaces suggested that the fraction of particles at the surface was insufficient. Rather than increasing the volume fraction of SiC in the material, which would significantly increase the material cost, Volvo used an etching process to remove aluminum from the surface, thus exposing a higher concentration of SiC. The effect on the SiC surface concentrations (in black) can be seen for the unetched and etched conditions in Figures 2 and 3, respectively.

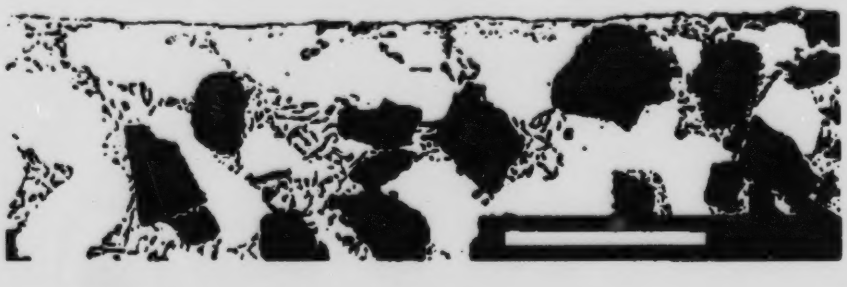

Figure 2 Typical Unetched MMC Disc Cross Section

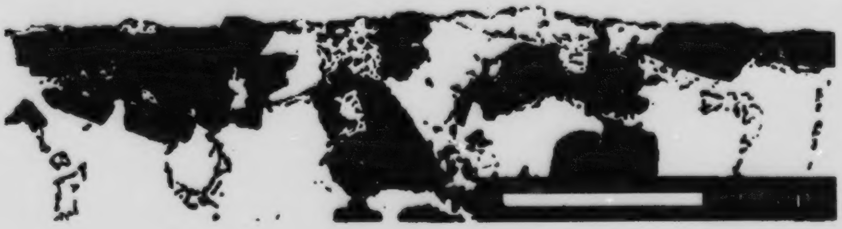

Figure 3 Surface Etched MMC Cross Section

An important requirement for an MMC brake is the ability to create a stable transfer layer without any special bedding-in procedures. Volvo found that the non surface-treated discs did not fulfil this requirement. Tests showed that these discs should not be exposed to any heavy braking before the transfer layer has been developed, which could take 20 to 50 moderate brake applications, depending on the lining formulation.

For the treated discs there was no need for a special bedding-in procedure. The number of brake applications needed for the development of the transfer layer on the treated disc was typically one or two. This was seen as the effect of the treatment in increasing the surface roughness. The topography is quickly smoothed during the first brake applications when the transfer layer is being developed, shown in Figure 4.

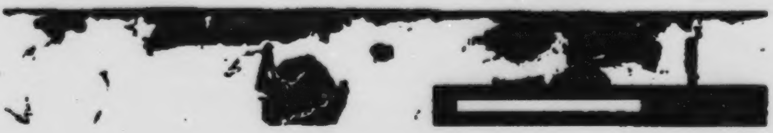

Figure 4 Etched Disc Cross Section After First Brake Application

The increased roughness of the disc surface provided a fast buildup of a robust and stable transfer layer. The rough interface between the disc surface and the transfer layer increases the adhesion of the transfer layer and thus contributes to the increased temperature stability.

Volvo found that a maximum disc temperature of 450°C could be permitted during severe acceptance testing. The maximum surface temperature was considerably higher, but caused no particular damage on surface-treated discs.

Because of the high thermal conductivity of the MMC material, the influence of the wheel on the disc temperature was found to be considerable. Tests in the Arizona desert with the Volvo S40 showed that the front disc temperature decreased about 100°C in a stop and go test with aluminum wheels as compared to steel wheels.

The Volvo developments provide further indication that MMC can be made to physically survive the high thermal stress of vehicle applications. As the specific energy input is higher in the automotive disc application than in the truck drum, it would be anticipated that the peak temperatures would be lower in a truck application. No modelling of the Volvo design was undertaken, but it is judged that the addition of the surface etching will reduce the peak surface temperatures predicted by the NRCan model.

Cost Analysis

This part of the study developed cost models and analysed the manufacturing costs for a variety of materials and manufacturing methods. Following is a list of the models that were developed and used for analysis of alternative materials for brake rotors and drums:

- Cast iron drum
- Cast iron rotor
- NRCan drum – with preform infiltration
- NRCan rotor – with preform infiltration
- MMC drum – from MMC billet

- MMC rotor – from MMC billet
- MMC drum – with MMC mixed on site
- MMC rotor – with MMC mixed on site

For consistency, certain global assumptions were made that would apply to each process/material choice:

- A production volume of 200,000 parts per year was chosen as a reasonable market share that would reflect the potential gap between current cost factors in cast iron parts and the alternatives.
- Certain input factors, such as labour rates, depreciation times for equipment and buildings, and energy rates, were held constant throughout the analysis.
- One rotor design and one drum design were applied to all materials and processes.
- No accommodation was made for change in material properties or potential design changes that might be necessary or made available.
- The energy required for melting the aluminum and MMC, the holding times required, and the lade travel times were all used to calculate the energy required in the production of parts in a separate model.

In this study the manufacturing costs for brake components of class 7/8 vehicles were evaluated for drums and rotors of several materials and manufacturing processes. The particular cases evaluated are shown in Table 2.

Table 2 Materials/Manufacturing Methods and Weights of Components

Material/Manufacturing System	Component Weight	
	kg	lb.
Cast iron drum	50.8	112
Cast iron rotor	27.2	60
NRCan drum – with preform infiltration	21.3	47
NRCan rotor – with preform infiltration	11.8	26
MMC drum – from MMC billet	21.3	47
MMC rotor – from MMC billet	11.8	26
MMC drum – with MMC mixed on site	21.3	47
MMC rotor – with MMC mixed on site	11.8	26
Carbon/Silicon Carbide ceramic rotor	10	22

For each material/technology evaluated, a cost model was prepared using input variables, specific material costs, recoveries and rejection rates, equipment costs and capacities, labour requirements, and energy consumptions, among others. Generally, these models follow the process route in the manufacture of the component. A sample process flow diagram, cost summary, and sensitivity analysis are shown in Table 3 and in Figures 5 and 6. It should be noted that all costs are presented in US dollars.

The material costs were calculated through part size, recoveries and rejection rates within each process step. By determining the number of direct operators for each process/machine and indirect labour for the manufacturing facility, a total labour cost was calculated and distributed over the production volume. Likewise the capital investment, energy costs, and other costs involved in the manufacture of the component were calculated, as was the sensitivity to changes in input variables.

Table 3 Cost Summary for PM MMC Drum from Billet

DURALCAN MMC BRAKE DRUM - CLASS 7/8 TRUCK
Permanent Mould, MMC from Billet

		Cost per part (US$/part)	**Cost per year** (US$/year)	**Investment**
Variable Costs				
	Materials	$67.94	$13,588,203	
	Direct Labour	$3.81	$762,048	
	Energy	$2.28	$455,768	
	Total Variable Cost	$74.03	$14,806,019	
Fixed Costs				
	Equipment	$1.32	$264,333	$3,965,000
	Tooling	$0.24	$48,040	
	Building	$1.25	$250,000	
	Indirect Labour	$0.95	$189,000	
	Total Fixed Cost	$3.76	$751,373	
	GRAND TOTAL	**$77.79**	**$15,557,392**	

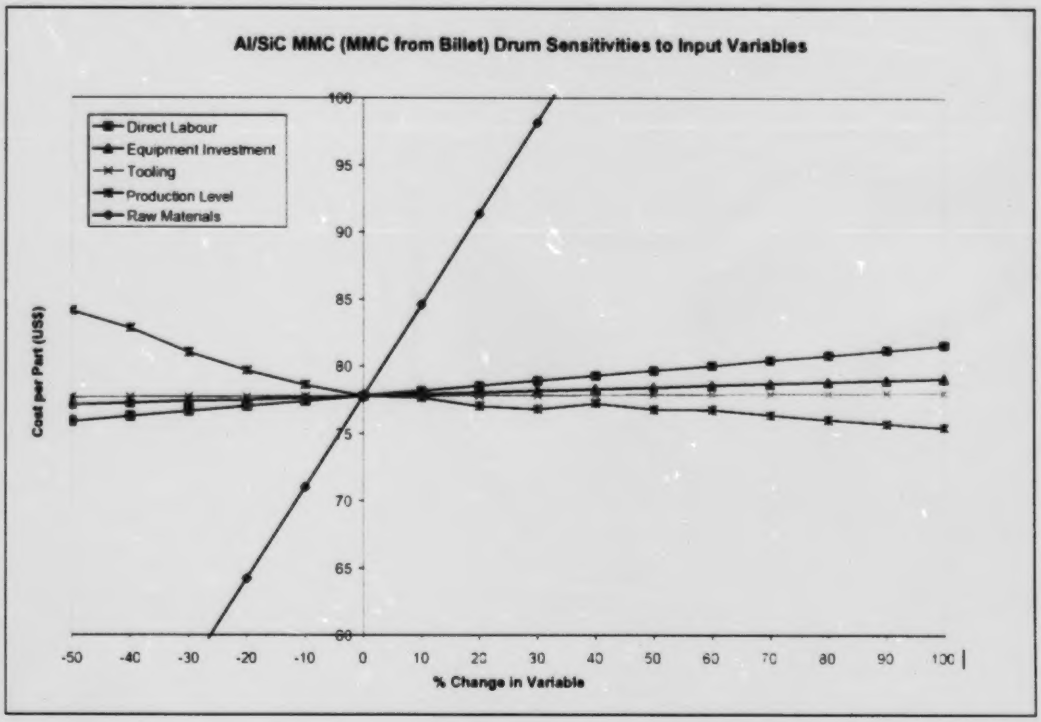

Figure 5 Sensitivity Chart for PM MMC Drum from Billet

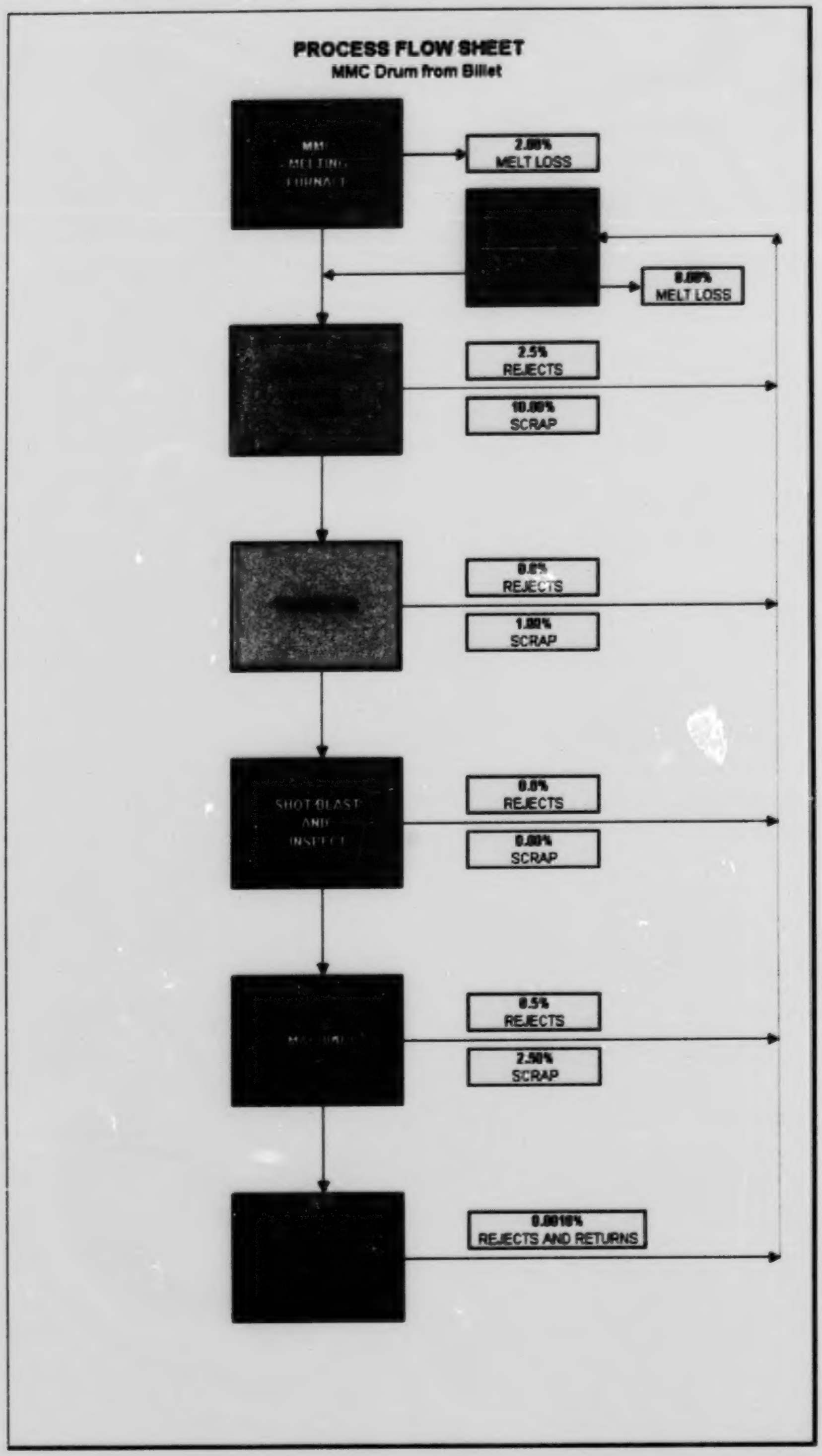

Figure 6 Process Flow Diagram for PM MMC Drum from Billet

Table 4 Materials/Manufacturing Methods and Costs of Manufacture

Material/Manufacturing System	Cost of Manufacture/Unit		Unit Weight		C$/kg	Cost/ Wt Saved (C$/kg)	
	US$	C$	lb.	kg		Drum	Rotor
Cast iron drum	$32.2	$48.8	112	50.8	$1.0		
Cast iron rotor	$31.8	$48.2	60	27.2	$1.8		
NRCan drum – with preform infiltration	$56.9	$86.2	47	21.3	$4.0	$1.3	
NRCan rotor – with preform infiltration	$40.8	$61.8	26	11.8	$5.2		$0.9
MMC drum – from MMC billet	$77.8	$117.9	47	21.3	$5.5	$2.3	
MMC rotor – from MMC billet	$46.7	$70.7	26	11.8	$6.0		$1.5
MMC drum – with MMC mixed on site	$57.0	$86.3	47	21.3	$4.0	$1.3	
MMC rotor – with MMC mixed on site	$35.8	$54.3	26	11.8	$4.6		$0.4
Carbon/Silicon Carbide ceramic rotor	$156.0	$236.4	22	10.0	$23.6		$10.9

The results of the manufacturing cost analyses for select drum and rotor designs for the various materials/technologies modelled are summarized in Table 4. In all cases, the cost is most sensitive to the raw material cost.

The analysis shows that none of the MMC materials had lower costs (either total or on a cost-per-kilogram basis) compared to the cost of manufacturing the components in cast iron. This is especially clear in the case of the drum, a part with a greater raw material requirement and fewer casting details.

The price penalty for the materials is significantly less if the cost per mass saved – a number used by the market – is calculated as shown in the last two columns of Table 4. For the drum configurations, the lowest incremental costs are achieved (C$1.3/kg) either with the preform or MMC mixed on site models. The premixed MMC is significantly higher at C$2.3/kg. For rotors, the costs are lowest ($0.4/kg) for the mix on site option. The preform method is estimated at slightly over twice this cost at C$0.9/kg and the premixed MMC would cost C$1.5/kg. Finally the C/SiC material has the highest cost at C$10.9.

In all cases, the highest degree of sensitivity belongs to the raw material cost, meaning that the amount required per finished part will most significantly influence the per unit cost. The use of infiltrated preform offers some advantage in terms of machining cost and, in the case of the larger, more material-intensive drum part, is estimated to be approximately equal in the cost of manufacture to the drum with an integrated MMC mixing/scrap reprocessing process. The integrated MMC mixing process is less costly in the manufacture of the rotor part.

These costs are at the "factory gate" and must be inflated to obtain estimates of the retail price differentials. There are a variety of factors that affect the size of the inflator, but it is generally in the range of 2 to 2.5 for automotive parts. Thus, many of the material options in the drum configuration come close to meeting the C$3/kg target required for wide market penetration.

Concluding Remarks

The potential for MMC materials to be used in truck brake drums and rotors can be assessed based on two criteria – performance and price. The performance criterion was assessed by the analysis of the physical properties and performance of the three candidate materials – MMC, CANMET composite and C/SiC. The criteria are mutually exclusive (i.e., both criteria must be met for the product to be viable).

The conclusion of the physical analysis indicated that the aluminum-based materials would be applicable for truck drum brake applications but not for truck disc brake applications. The high temperature of the disc brakes was determined to exceed the MOT of the two aluminum materials. The C/SiC was assessed to be applicable for both configurations.

The market analysis indicated that over 90 percent of the current trucks use drum brakes but that the use of disc brakes is expanding rapidly in Europe. This trend may extend to North America but probably not without a strong regulatory push or significant price advantages. The market, being very sensitive to product price, will pay a premium for weight or maintenance cost reductions. The analysis estimated that a high penetration of both the new vehicle market and aftermarket could be expected for lighter weight brakes if the cost differential was below C$3/kg. The market acceptance drops precipitously after this cost range but there are segments of the market that will pay high differentials. For example, it is estimated that approximately 20 percent of the market would accept a cost differential as high as C$10/kg.

The results of the manufacturing cost analyses for select drum and rotor designs for the various materials/technologies modelled estimated that:

- for the drum configurations, the lowest incremental costs are achieved (C$1.3/kg) either with the preform or MMC mixed on site models;
- the premixed MMC is significantly higher at C$2.3/kg;
- for rotors, the costs are lowest ($0.4/kg) for the mix on site option;
- the preform method is estimated at slightly over twice this cost at C$0.9/kg and the premixed MMC would cost C$1.5/kg;
- the C/SiC material has the highest cost at C$4.6/kg.

It is thus concluded that the use of aluminum MMC – either in homogeneous mixture with SiC or as a composite with Al_2O_3 – has a high probability of being both physically and financially viable for the drum brake market.

Development Requirements and Strategy

The successful further development of heavy truck brake drums and rotors requires a large number of connected factors and resources. This section will describe these requirements and lay out a possible development strategy for the commercialization of heavy-duty brakes made of the new materials.

There are three major issues that must be dealt with:

1. The production of new prototype drums and their testing, which will ensure that the results of the physical modelling are accurate and that the product will sustain the thermal and stress environments of a heavy-duty truck or bus.
2. The strategic connection with a drum manufacturer, which will ensure that the product, if it passes the technical tests, can be introduced to the market in an efficient and effective manner.
3. Confirmation of the financial models. After the technical evaluation has been successful, more detailed analysis of the production cost options must be reviewed to ensure that the product cost base as modelled is correct.

Physical Performance Testing

A series of steps must be undertaken to develop, test and qualify the feasibility of using MMC or the NRCan composite material in a drum brake. The testing must validate, first in controlled laboratory testing and then on the test track, the physical reliability and durability of the MMC-based drum product.

For the MMC material, the material, casting techniques and casting facilities are already in existence. However, for the NRCan composite material, the final material must be formulated and tested for its physical characteristics to confirm the estimations used in the analysis model. It must then be cast by NRCan using a custom-built squeeze cast mould. Thus, two distinct evaluation paths are described in the following text.

The MMC-based design can be based on the material, casting and finishing techniques used previously by Hayes Lemmerz and Volvo. The most expeditious route for this option is to work in collaboration with the companies and suppliers that have been involved in the previous efforts and thus this is the focus of this path. The work steps are:

1. Develop Drum Design
2. Cast and Prepare Drum
3. Select Friction Pad Material
4. Conduct Brake Dynamometer Testing
5. Conduct Test Track Evaluation
6. Complete Evaluation

The evaluation of the NRCan composite material contains a number of additional steps. NRCan must undertake the work because of the proprietary nature of its material and processes. The work steps to create prototype drums are outlined in the following text. After the delivery of the prototype drums, the test procedures from MMC Task 3 on would be followed.

1. Fabricate Composite Test Samples
2. Evaluate Composite Test Samples
3. Select Drum Brake Rotor Type and Fabrication Process
4. Fabricate Squeeze Cast Die
5. Fabricate Squeeze Cast Composite Drum Brake Rotors
6. Analyse Data and Report Results

Test Program Costs and Schedule Estimates

The MMC-based prototype casting and testing program is estimated to cost C$100,000 (see cost breakdown in Table 5). Roughly half of this total budget is the direct costs associated with casting, finishing, and testing or material analysis of the drums. The balance of the budget covers the costs of project management and travel associated with co-ordinating, monitoring and reporting on the test program.

The estimated project schedule is summarized in Figure 7. The time required for the castings is quoted as three weeks and the CMVSS/FMVSS 121[2] tests are normally completed in a two-week period. The extended track testing is estimated to require an additional four weeks. The total testing program is estimated to require six months to complete.

The NRCan product development program to develop and deliver six composite and six MMC prototypes is estimated at C$120,000 (see Table 6) and will require approximately 12 months to complete. After prototype delivery, there will be an additional C$88,000 (and five months) required for the lab and test track evaluation program (C$100,000 minus the direct costs of the MMC casting C$12,000) for a total of C$208,000.

Table 5 MMC Drum Testing Program Cost Breakdown

		Costs by Task (C$)					
Task	Title	Prj Mgr	Sub-Total Labour	Expense			Project
	Per Diem Fees (8 h/day)	$1,120		Travel	Misc.*	Sub-Total	Total
1	Finalize Testing Plan, Participants and Suppliers	$6,720	$6,720	$4,480		$4,480	$11,200
2	Cast Drums (12)	$0	$0	$0	$10,800	$10,800	$10,800
3	Machine and Finish Drums	$0	$0	$0	$1,500	$1,500	$1,500
4	Select Friction Material	$4,480	$4,480	$2,240	$2,000	$4,240	$8,720
5	Conduct Laboratory Brake Testing	$0	$0	$0	$7,200	$7,200	$7,200
6	Analyse Material	$3,360	$3,360	$0	$2,000	$2,000	$5,360
7	Test Track – CMVSS/FMVSS 121	$5,600	$5,600	$2,600	$12,000	$14,600	$20,200
8	Test Track - Extended Tests	$3,360	$3,360	$2,360	$15,000	$17,360	$20,720
9	Evaluate Material and Performance	$3,360	$3,360	$0	$3,000	$3,000	$6,360
10	Produce Report	$4,480	$4,480	$0	$500	$500	$4,980
	Total	**$31,360**	**$31,360**	**$11,680**	**$54,000**	**$65,680**	**$97,040**

* Misc. includes communications, courier, printing and other directly incurred costs.

10%	Contingency	$3,136
	Total	$100,176

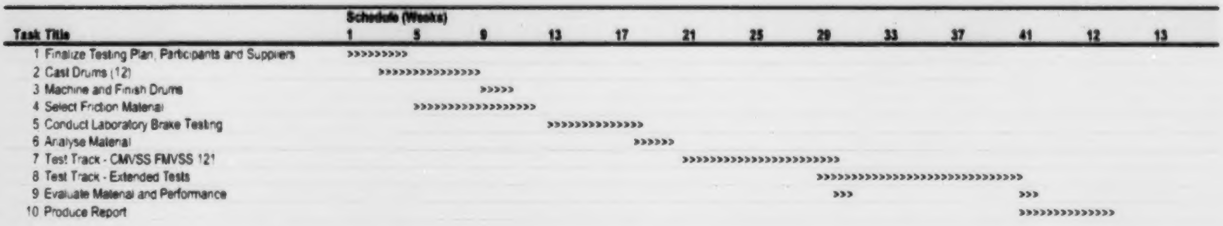

Figure 7 Test Program Schedule

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[2] CMVSS/FMVSS 121 is the identical US and Canadian federal regulation establishing the design and performance requirements of air brake systems for motor vehicles. As part of the regulation, a vehicle is required to perform a dynamic braking test consisting of 10 consecutive stops from 80.5 to 24.2 km/h at a deceleration rate of 2.72 m/s², followed by a stop from 32.2 to 0 km/h at a deceleration rate of 4.27 m/s².

Table 6 Cost Breakdown for NRCan Composite Brake Drum Fabrication

Major Task	SubTotal (C$)
Fabricate Composite Test Samples	$10,000
Evaluate Composite Test Samples	$15,000
Select Drum Brake Rotor Type and Fabrication Process	$2,000
Fabricate Squeeze Cast Die	$30,000
Fabricate Squeeze Cast Composite Drum Brake Rotors	$45,500
Analyse Data and Report Results	$5,000
Materials & Supplies	$8,000
Travel	$3500
Total	**$120,000**

Product Development Partner Options

In order for a successful product to be developed and launched into the heavy-duty brake market, a number of key actors must be associated with the development project. These include:

Product Manufacturer/Distributor

As noted in the market analysis, the brake drum industry consists of a handful of OEMs that either supply just the drums or integrate the drum into a complete brake system, which is sold to the vehicle OEM or to the aftermarket. One of the OEM's primary roles is to accept the design responsibility and liability of the product. This is a substantial obligation and makes entry into the market by a start-up company difficult, if not impossible. Also, the OEM has developed and controls a market distribution chain of distributors, jobbers, etc. Again, this parts marketing and distribution system takes a long time and substantial financial investment to create, making it essential for any new product to be developed in conjunction with a major OEM such that the new product can enter the market with as much technical credibility and market impetus as possible.

During the study, the project team met or interviewed most of the current OEMs supplying the North American market. The most recent experience with MMC was found to be at Hayes Lemmerz which, while having "shelved" its product development, was open to collaboration on a revised design and material for its MMC brake drum. At present, the company is willing to participate in an evaluation project and could contribute its existing mould patterns, laboratory test time and technical support to the project.

Casting Material Supplier

There are a number of potential suppliers of MMC, including Alcan, Norsk Hydro and MC-21. At the proof of concept stage it is relatively unimportant as to which of the North American-based suppliers' material is used (Duralcan or MC-21). Based on Volvo's experience, it appears physically possible and possibly better to use a SiC particle size mixture that is broader than that normally used in the Duralcan product. The advantage of the broader size spectrum is a reduced product cost (of no importance at the evaluation stage) and possibly increased matrix strength and thermal transfer properties. The downside of the larger particles is that proper mixing of the particles and keeping them in suspension during the casting process is more problematic.

Preliminary discussions with both Duralcan and MC-21 have proved positive with both expressing interest in the evaluation program.

Caster

The choice of caster for the MMC material should be based on prior familiarity with MMC casting and heavy-duty drum designs. The previous casting of the Hayes Lemmerz drum was done by Eck Foundry and, for convenience and continuity, any new casting should be done with Eck. Other Ontario and Quebec aluminum casting plants have the capabilities to produce pilot or full production quantities when that stage is attained. Sourcing for the production casting is not considered a priority in this evaluation stage of the product development.

Friction Material Supplier

The other side of the friction couple is a key factor in the success of the MMC drum. The formulation of the pad material for good friction layer development, friction coefficients and wear is a difficult task. A number of friction material suppliers have been involved in MMC brake programs in the past and all are likely to be interested in participating in any heavy-duty program. None of the friction material suppliers are located in Canada.

Vehicle OEMs

The merits of lighter brake systems are well understood by the vehicle OEMs and a number of the larger truck manufacturers expressed interest in the potential product. The most energetic response was obtained from Oshkosh Trucks (Wisconsin), which stated that it wished to participate in any trial of the brake drums. Oshkosh saw an immediate application in its cement mixer product lines and possibly in its military vehicles. No other OEM (bus or truck) was this positive in its consideration of the product and evaluation project.

The selection of the initial test vehicle is somewhat unimportant except that it should represent a Class 8 tandem axle truck. The ability to use the vehicle to simulate an urban bus application would be beneficial and could be achieved through either disablement of the second rear brake set or the availability of a bus test vehicle.

End User

For the prototype test and evaluation stage, it is too early to heavily involve end users. However, the development of a user group that could provide test and brake performance requirements advice to the program would be useful. The existence of this advisory group would also accelerate the development of a field demonstration, assuming that the product is found to be satisfactory in the controlled tests. The user group should include staff from severe duty application fleets (cement mixers, dump trucks, etc.) and transit properties.

Canadian Content Issues

The brake manufacturing industry is controlled by a handful of multi-national companies, none of which are based in Canada. While there are casting companies located in Canada that provide brake components, most of these companies supply to the light-duty market. There are, at present, no known suppliers for heavy-duty brake drums (or discs) in Canada. There are a number of aluminum casters that have the capabilities to cast MMC (or the NRCan composite material) and these companies could be developed into brake part suppliers if the product is commercialized.

For the prototype testing phase, most of the resources and partners are non-Canadian. Parts of the work, such as project management and track testing, could be Canadian sourced. Also, it is probable that commitments to Canadian sourcing of casting or materials could be negotiated based on provision of Canadian government financial investments in the further product testing and development phase.

Recommendations

Based on the analysis undertaken in this study, it has been concluded that the use of MMC materials in heavy-duty vehicle brake drums should be both physically feasible and financially attractive. However, the physical performance has not yet been demonstrated and thus it is recommended that a prototype design, manufacture and test program be undertaken in an attempt to demonstrate the physical performance of the material.

The two candidate material forms (MMC and NRCan composite) require different investments of resources to complete preliminary performance tests of the prototype designs. The analysis of the prototype testing costs and time schedules indicate that the MMC-based product would be less expensive and faster than the NRCan composite material, and have a lower development risk.

It is recommended that:

- A program should be developed in conjunction with a brake drum manufacturer to undertake the MMC-based drum test and evaluation program outlined above. Based on early discussions, substantial funding support could be provided by the brake drum OEM and friction material suppliers.
- Because the NRCan design has significant technical and economic merit, and if additional funding were available, it should also be supported as it potentially provides a less expensive manufacturing process and could result in higher Canadian content because of the intellectual property rights that NRCan has in the process.
- Based on the successful completion of a test of the prototype material(s), a larger demonstration of the product should be developed and more detailed production planning and manufacturing cost analysis be undertaken.

SOMMAIRE

Cette étude, commandée par le Centre de développement des transports de Transports Canada, a consisté en une vaste étude technique et une analyse du marché concernant l'utilisation de composites à matrice métallique (CMM) légers à base d'aluminium dans les tambours et disques de freins pour véhicules lourds. Comme les systèmes de freinage comptent pour une large part des coûts d'entretien des parcs de véhicules, en particulier des véhicules de transport en commun, le recours à des matériaux de pointe plus légers, à usure plus lente et nécessitant moins d'entretien que les matériaux courants serait très avantageux au triple chapitre des coûts, de la sécurité et de la consommation de carburant.

Les matériaux auxquels s'est particulièrement intéressée cette étude sont les CMM à matrice d'aluminium. Cette matrice existe sous deux formes. La première est un composite homogène prémélangé, soit un alliage d'aluminium renforcé de carbure de silicium (SiC/Al), développé par Alcan Ltd. et commercialisé sous le nom de Duralcan[1]; la deuxième est un composite non homogène (de structure dite sandwich) dans lequel l'aluminium est infiltré dans une préforme d'éléments de renfort, habituellement de l'alumine (Al_2O_3). Ce dernier matériau découle d'une technique brevetée par le Laboratoire de technologie des matériaux de Ressources naturelles Canada (RNCan). Dans les deux cas, le but est d'incorporer à l'aluminium un agent de renforcement et de transfert de chaleur, de façon à pouvoir élargir la gamme de ses applications, notamment dans des conditions générant de fortes contraintes et des températures élevées.

La proportion de matériaux légers utilisés dans les véhicules augmente au fur et à mesure que s'intensifie le besoin de réduire la consommation de carburant et d'augmenter les performances des véhicules, et que diminuent les coûts de fabrication et d'application des matériaux de substitution légers. La quantité de matériau léger utilisée dans les véhicules de transport est directement proportionnelle à l'importance que le constructeur ou l'utilisateur final accorde à l'allégement du véhicule. C'est ainsi que les avions sont largement composés de matériaux légers (aluminium, titane, plastiques, etc.). Les véhicules routiers contiennent une proportion plus faible de matériaux légers, plus coûteux que les matériaux classiques, proportion qui, là encore, dépend de leur vocation et du marché qu'ils desservent.

Depuis toujours, les tambours et disques de freins pour véhicules légers et véhicules lourds sont faits de fonte. Le grand avantage de la fonte est qu'elle est abondante et bon marché. Mais elle a des inconvénients : son poids, qui mine l'efficacité énergétique des véhicules; ses piètres propriétés physiques, qui engendrent bruit, vibrations et freinage saccadé; et son taux d'usure relativement élevé, comparé à celui de certains matériaux de substitution envisagés, qui oblige à des réglages fréquents.

Grâce au développement de nouveaux composites affichant des propriétés améliorées, les véhicules légers sont maintenant équipés de disques de freins en composite (qui parent la plupart des inconvénients associés aux disques en fonte). La présente étude évalue les chances d'étendre avec succès l'utilisation de disques de freins en composite aux véhicules lourds. Elle cherche plus précisément à savoir si des tambours et disques de freins en composite peuvent répondre aux critères de performance et de fabrication et aux autres critères d'entrée sur le marché.

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[1] D'autres producteurs d'aluminium, comme Norsk Hydro, ont aussi élaboré des CMM semblables.

L'étude comprend quatre grandes étapes :

1. Revue de la technologie des freins et du marché actuel des composants de freins pour véhicules lourds, et estimation de la valeur économique potentielle des caractéristiques (poids et taux d'usure) des nouveaux matériaux envisagés pour la fabrication de freins.

2. Analyse des matériaux de substitution et de leur aptitude technique à être utilisés dans les tambours et disques de freins pour véhicules lourds.

3. Modélisation des coûts de production de freins en matériaux de substitution.

4. Analyse des chances de succès technique et économique d'un tambour ou d'un disque de frein utilisant des matériaux de substitution, et élaboration d'un plan pour assurer ce succès.

Les conclusions de chaque étape sont résumées ci-après.

Analyse du marché

Les sollicitations exercées sur les systèmes de freinage ont énormément augmenté ces 50 dernières années. On estime, en effet, que la puissance de freinage exigée des systèmes a augmenté de 30 p. 100 en raison de :

- l'augmentation des vitesses permises,
- la hausse des limites de poids nominal brut des véhicules (PNBV),
- la diminution de 30 p. 100 de la décélération naturelle des véhicules, par rapport à il y a 15 ans,
- la diminution de 19 p. 100 de la puissance de freinage du moteur,
- la mode des véhicules surbaissés.

L'énorme force de freinage nécessaire pour arrêter un camion lourd génère une quantité de chaleur tout aussi énorme. Les freins d'un camion sont évidemment beaucoup plus gros, en termes absolus, que ceux d'une voiture particulière. Mais en proportion du poids du véhicule, ils sont plus petits. Plus précisément, la surface de contact du tambour (la «superficie balayée») est considérablement plus petite (deux à quatre fois plus petite, par rapport au poids du véhicule) dans les ensembles articulés à système de freinage pneumatique que dans les voitures particulières, comme le montre le tableau 1. Comme les freins des camions sont relativement plus petits (par rapport au poids du véhicule), il s'ensuit que leur capacité thermique est moindre que celle des freins des voitures particulières. Aussi, comme les forces de freinage sont beaucoup plus grandes, la puissance spécifique qu'ils ont à absorber peut atteindre quatre fois celle absorbée par les freins d'une voiture.

Tableau 1 Comparaison des freins à disques des voitures particulières et des camions

Type de véhicule	*Superficie de la garniture de frein (cm²/kg du PNBV)*	*Superficie balayée (cm²/kg du PNBV)*	*Puissance spécifique (kW/m²)*
Voiture particulière	0,40	1,21	67
Tracteur semi-remorque	0,36	0,57	300
Rapport	0,9	0,5	4,5

Les systèmes de freinage se divisent en deux grands types : les freins à tambours et les freins à disques. Avant les années 1970, des freins à tambours équipaient les roues avant et arrière de la plupart des voitures et des camions. Ce type de frein est encore très courant sur les roues arrière des voitures de série et sur tous les véhicules commerciaux, sauf en Europe. Les freins à disques ont beaucoup augmenté leur part du marché des voitures particulières. En Europe, la plupart des camions et remorques neufs sont équipés de freins àdisques.

La valeur totale du marché des tambours et disques de freins est évaluée àprès de 1 milliard $US, tant en Amérique du Nord qu'en Europe. Le marché des composants de freins est approvisionné par des équipementiers, qui souvent ont eux-mêmes recours à des fournisseurs, ou par des distributeurs de pièces détachées. En Amérique du Nord, tous les équipementiers font affaire avec des fournisseurs tiers de composants de freins. En Europe, les équipementiers fabriquent environ 50 p. 100 de leurs composants, mais ce pourcentage diminue rapidement au profit des fournisseurs de premier rang. Cela dit, le marché des freins et composants est dominé par un petit nombre d'entreprises. À titre d'exemple, le plus gros fournisseur, ArvinMeritor, occupe environ 80 p. 100 du marché des pièces pour véhicules neufs en Amérique du Nord. Dans le cas des pièces détachées, les fournisseurs sont plus nombreux, mais le marché des tambours (plus de 80 p. 100) est dominé par trois ou quatre entreprises (ArvinMeritor, Webb Wheels, Hayes Lemmerz, Gunite). En raison de la taille et de la compétitivité du marché des tambours de freins, la grande majorité des fournisseurs de premier rang achètent les tambours bruts àdes fonderies, se contentant ensuite de tester, usiner, emballer et distribuer le produit.

Les marchés nord-américain et européen des composants de freins diffèrent àplusieurs égards :

- En Amérique du Nord, les acheteurs de camions ou de remorques précisent, au moment de la commande, les composants (ou le fournisseur) du système de freinage, tandis que dans l'Union européenne (UE), toutes les décisions concernant le système de freinage reviennent àl'équipementier.

- Les normes qui régissent les freins dans l'UE sont différentes des normes nord-américaines. Ce sont essentiellement ces normes, en exigeant des freins capables de dissiper une plus grande énergie, de réagir plus rapidement etc., qui ont favorisé l'essor des freins àdisques en Europe.

- Comparativement àl'UE, l'Amérique du Nord compte un plus grand nombre de parcs de véhicules plus petits, ce qui tend à disperser les achats. De plus, les ressources techniques affectées àla spécification ou àl'essai de produits sont moins nombreuses.

Les différences entre les réglementations américaines (et canadiennes) et européennes en ce qui a trait aux performances des nouveaux produits expliquent en grande partie le virage observé sur le marché de l'UE vers les freins à disques et les systèmes de freinage électroniques. En effet, les normes de performances édictées par l'UE sont davantage axées sur le freinage haute énergie, la stabilité du freinage et l'évanouissement (*fading*) que les règles américaines. Or, les freins à disques sont mieux en mesure de répondre à ces exigences, parce qu'ils sont, par nature, moins sujets à l'évanouissement et que leur temps de réaction est plus court. Cette pression réglementaire, conjuguée aux bases de produits différentes des fournisseurs de tambours et de disques en Europe, a remodelé, ces cinq dernières années, le marché des freins pour camions neufs : environ 80 p. 100 des tracteurs et des camions porteurs, et 50 p. 100 des remorques sont maintenant équipés de freins àdisques. En revanche, en Amérique du Nord, les disques comptent pour moins de 1 p. 100 du marché des poids lourds. Il est permis de penser que seule une réglementation plus exigeante relativement aux performances des systèmes de freinage pourrait entraîner une percée majeure des freins à disques sur le marché nord-américain.

Une fois que les propriétés techniques des nouveaux matériaux pour freins auront été démontrées, le marché se montrera ouvert àla technologie des freins légers, en raison des économies directes qu'elle fera réaliser aux exploitants, économies surtout attribuables à l'allégement du véhicule. Selon les données actuelles du marché, environ 20 p. 100 des acheteurs sont très sensibles à l'argument «poids» et ils sont prêts à débourser jusqu'à 10 $CAN pour alléger d'un kilogramme leur véhicule. Dans certains créneaux, on est prêt à payer encore plus. S'il était possible d'abaisser le coût différentiel à 3 $CAN/kg, une percée importante des freins légers serait à prévoir sur le marché de masse des camions.

Le taux d'usure beaucoup plus faible des CMM et des composites carbone/carbone constitue donc un atout très important. Une étude des effets d'un abaissement du taux d'usure dans le cas d'un camion (ou d'un autobus) à un ou deux essieux moteurs a révélé qu'en réduisant de 25 ou 50 p. 100 le taux d'usure (ce qui revient à allonger de 50 à 100 p. 100 le délai de remplacement), on pourrait diminuer de 80 et 83 p. 100 le risque d'un mauvais réglage des freins. En comparaison, les dispositifs automatiques de rattrapage du jeu (qui font maintenant obligatoirement partie de l'équipement standard) réduisent de 75 p. 100 le risque de déréglage. Cette diminution du taux d'usure compense largement les coûts supplémentaires potentiels associés au changement de matériau (l'usure serait de 2 à 10 fois plus lente) et donne fortement à penser que l'utilisation de matériaux à faible taux d'usure pourrait grandement améliorer, dans l'ensemble, la performance en freinage.

L'effet de l'allégement potentiel du véhicule sur la consommation de carburant est fonction du cycle de service du véhicule et de sa configuration. Selon des données empiriques et les résultats d'études théoriques, c'est dans les véhicules exploités sur de longues et moyennes distances que l'allégement aurait le moins d'effet sur la consommation de carburant (1,2 point de pourcentage par 1 000 kg d'allégement). Quant aux camions porteurs utilisés pour les livraisons intra-urbaines et aux autobus de transport en commun, leur consommation de carburant pourrait être réduite de 2,4 et 3,4 points de pourcentage, respectivement, pour chaque 1 000 kg retranché.

Propriétés physiques des matériaux

L'analyse des propriétés physiques des matériaux s'est faite en plusieurs étapes :

1. Recherche documentaire sur l'essai et la modélisation de systèmes de freinage, et analyse des performances de matériaux CMM.

2. Revue des propriétés de base des matériaux en regard des performances en service exigées des composants de freins.

3. Revue des procédés de fabrication de composants de freins en composite pour véhicules légers.

4. Élaboration de modèles informatiques capables de prédire les profils de contraintes et de températures de disques/tambours de freins fabriqués à partir des matériaux candidats, dans des applications sur véhicules lourds.

Les composites possèdent des propriétés physiques, thermiques et mécaniques supérieures à celles des matériaux classiques, et sont plus légers. La faible température maximale d'utilisation (TMU) du SiC/Al, qui est de 450 °C, limite l'utilisation de ce matériau dans les freins. Le Al_2O_3/Al modifié par CANMET devrait afficher une TMU beaucoup plus élevée, de l'ordre de 600 °C, mais on ne dispose pas encore de tous les résultats des essais dynamométriques. Les composites carbone/carbure de silicium (C/SiC) ont affiché les meilleures performances, avec une TMU de 1 350 °C.

Une analyse de la fabrication a révélé qu'il est possible de mettre en forme tous les matériaux considérés aux cotes nécessaires. Ainsi, on a pu mettre en forme les deux composites d'aluminium au moyen de techniques de forgeage-moulage (compression à l'état liquide) plutôt que de moulage sous haute pression, ce qui est de nature à réduire les coûts de fabrication. Quant au composite C/SiC, il a servi à fabriquer des séries relativement limitées de diverses configurations de composants.

Les résultats de la modélisation informatique (àl'aide de valeurs extrêmement prudentes) réalisée par CANMET révèlent que la température des disques de freins en composite d'aluminium équipant des véhicules lourds peut atteindre 615 °C (la figure 1 donne un exemple de montée en température), et celle des tambours de freins, 447 °C. On peut donc raisonnablement conclure que si le SiC/Al convient àla fabrication de tambours de freins de véhicules lourds, il ne convient pas àla fabrication de disques. De même, les composites Al_2O_3/Al, élaborés par CANMET, peuvent soutenir les conditions de service rigoureuses imposées aux freins des véhicules lourds, pour autant que ces freins soient du type à tambour plutôt qu'à disque. Mais l'analyse des matériaux de CANMET présente un degré élevé d'incertitude, du fait que les résultats des essais de confirmation des propriétés des matériaux effectués sur les composites Al_2O_3/Al n'ont pas encore été divulgués.

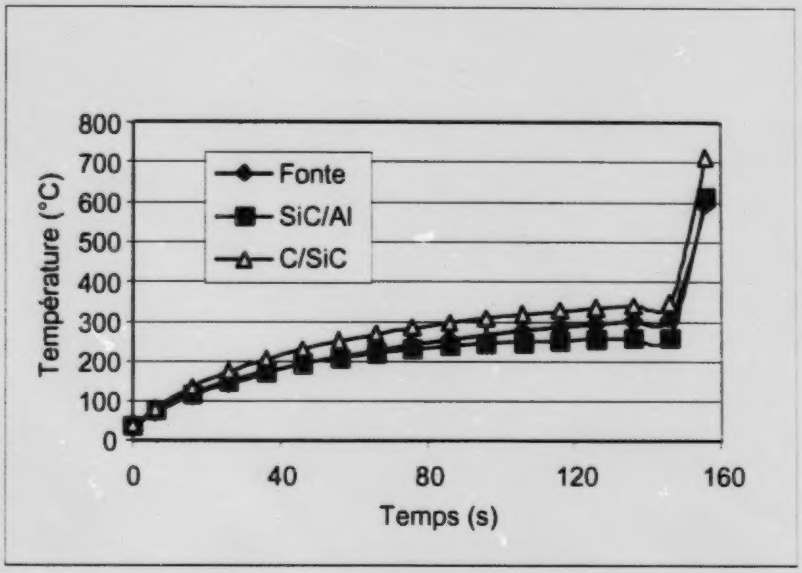

Figure 1 Modélisation informatique de la montée en température des disques de freins

Avec une TMU de 1 350 °C, le C/SiC est àcoup sûr le matériau qui affiche les propriétés physiques les plus intéressantes pour une utilisation dans des freins pour véhicules lourds. Quant à l'analyse des contraintes, elle a révélé que les trois composites sont en mesure de résister aux contraintes associées aux conditions de freinage les plus exigeantes.

Au cours de la présente étude, la société Volvo Technology Development Corp. a publié un rapport qui examine la possibilité d'améliorer la capacité thermique des CMM en augmentant légèrement la taille des particules de SiC comptant pour une fraction volumique de 20 p. 100 du composite à matrice d'aluminium, et en décapant la surface d'usure du produit fini. Le frein choisi pour les travaux de développement était le frein avant d'une voiture Volvo S40 équipée de roues de 15 pouces. Le diamètre a été imposé par la nécessité d'une substitution directe des disques en fonte existants. Ces travaux sont considérés très prometteurs pour l'utilisation future de CMM dans les freins.

Le bon fonctionnement d'un frein en CMM dépend de l'uniformité de la couche de transfert (interface entre le tambour ou le disque et les plaquettes). Volvo (et d'autres) ont constaté que le matériau dont sont faites les plaquettes influe sur la capacité de créer une telle couche de transfert. Par ailleurs, des recherches ont révélé que l'uniformité de la couche de transfert dépend non seulement du matériau dont sont faites les plaquettes mais aussi, et dans une grande mesure, de la structure de la surface du disque.

Lors d'essais réalisés par Volvo à l'aide de plusieurs plaquettes faites d'un même matériau, certains disques résistaient sans problème à des températures de plus de 450 °C, tandis que d'autres, malgré une répartition satisfaisante des particules, se rayaient à moins de 400 °C. Un examen de la surface des disques a révélé que la proportion de particules en surface était insuffisante. Mais plutôt que d'accroître la fraction volumique de SiC, ce qui aurait considérablement augmenté le coût du composite, Volvo a fait appel à un procédé de décapage chimique pour enlever de l'aluminium à la surface du disque et exposer ainsi une plus grande concentration de SiC. L'effet de ce procédé sur les concentrations de SiC en surface (en noir) est présenté aux figures 2 et 3, avant et après décapage, respectivement.

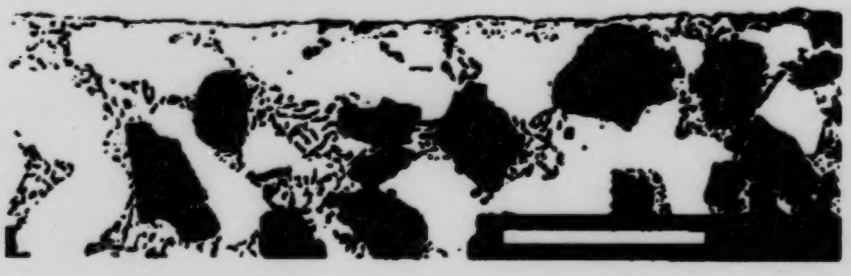

Figure 2 Vue en coupe d'un disque en CMM avant décapage

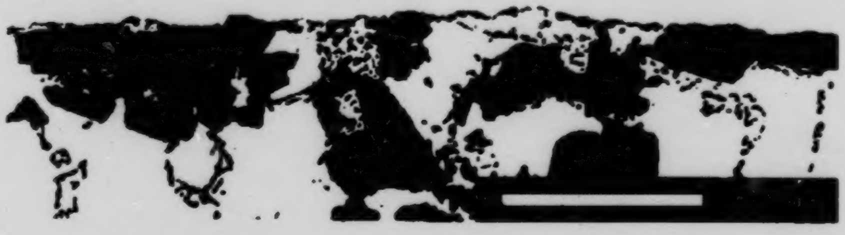

Figure 3 Vue en coupe d'un disque en CMM après décapage

Il est important, dans le cas d'un frein en CMM, de pouvoir créer une couche de transfert uniforme sans avoir à appliquer une procédure spéciale de rodage. Volvo a constaté que les disques qui n'avaient pas subi de traitement de surface ne remplissaient pas cette exigence. En effet, les essais ont révélé que pour exposer ces disques à un effort de freinage important, il fallait d'abord «créer» la couche de transfert, ce qui pouvait nécessiter de 20 à 50 applications modérées des freins, selon la formulation de la garniture.

Dans le cas des disques qui avaient subi un traitement de surface, aucun rodage spécial n'était nécessaire. Il suffisait habituellement d'appliquer une ou deux fois les freins pour créer la couche de transfert. Cela a été attribué au traitement de surface, qui augmentait la rugosité de celle-ci. Les aspérités de la surface sont rapidement abolies à la première application des freins, alors que s'établit la couche de transfert (voir la figure 4).

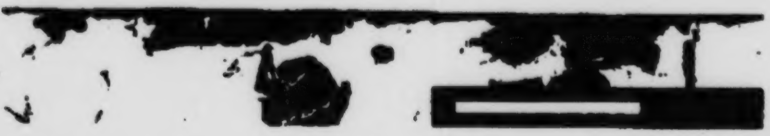

Figure 4 Vue en coupe d'un disque décapé, après une seule application des freins

La rugosité accrue de la surface du disque a permis d'établir rapidement une couche de transfert solide et uniforme. En effet, l'interface rugueuse entre la surface du disque et la couche de transfert favorise l'adhérence de cette dernière et contribue ainsi àstabiliser la température dans le disque.

Au cours d'essais d'acceptation rigoureux, Volvo a constaté qu'une température maximale de 450 °C pouvait être autorisée pour le disque. La température maximale atteinte par la surface était considérablement plus élevée, mais cela n'a pas causé de dommage particulier aux disques qui avaient subi un traitement de surface.

Comme les CMM présentent une conductivité thermique interne supérieure à celle des matériaux classiques, le type de roue exerce une influence considérable sur la température du disque. En effet, lors d'essais de type «arrêt-démarrage» menés dans le désert de l'Arizona avec la Volvo S40, les disques des roues avant affichaient une température d'environ 100 °C de moins, lorsque les roues étaient en aluminium plutôt qu'en acier.

Les travaux réalisés par Volvo confirment qu'il est réaliste de penser que les CMM possèdent les propriétés physiques pour résister aux fortes contraintes thermiques associées aux applications dans des véhicules. Comme l'apport d'énergie spécifique est supérieur dans un disque de frein pour voiture particulière que dans un tambour de frein pour camion, on peut penser que les températures maximales seraient plus faibles dans un camion. La technique de Volvo n'a fait l'objet d'aucune modélisation, mais on estime que le décapage chimique de la surface est de nature à réduire les températures maximales des surfaces prédites par le modèle de RNCan.

Analyse des coûts

Cette partie de l'étude a consisté à élaborer des modèles de coûts et à analyser les coûts de fabrication de composants de freins en divers matériaux, selon divers procédés de fabrication. Voici la liste des modèles mis au point et utilisés pour l'analyse :

- Tambour en fonte
- Disque en fonte
- Tambour en composite de RNCan – infiltration dans une préforme
- Disque en composite de RNCan – infiltration dans une préforme
- Tambour en CMM – billette de CMM
- Disque en CMM – billette de CMM
- Tambour en CMM – CMM mélangé sur place
- Disque en CMM – CMM mélangé sur place

Pour faciliter la comparaison, certaines hypothèses générales ont été posées pour chacune des options procédé/matériau :

- Le volume de production de chaque option a été établi à200 000 pièces par année, ce qui représente une part de marché raisonnable, compte tenu de la différence àprévoir entre le coût des pièces en fonte et des autres.
- Certains facteurs d'intrant, comme les coûts de main-d'œuvre, la dépréciation du matériel et des immeubles, et les coûts d'énergie, ont été maintenus constants.
- Les matériaux et procédés ont été comparés en regard d'un seul modèle de disque et d'un seul modèle de tambour.
- Aucun accommodement n'a été prévu pour le cas où il serait nécessaire ou possible de modifier les propriétés des matériaux ou la conception des pièces.
- Un modèle distinct a été utilisé pour calculer l'énergie nécessaire pour produire les composants (fonte de l'aluminium et du CMM, temps de retenue nécessaires, temps de déplacement de la poche de coulée).

La présente étude a porté sur les coûts de fabrication de composants de freins pour véhicules de classes 7/8, soit des tambours et des disques faits de différents matériaux, àl'aide de divers procédés. Les cas particuliers évalués sont montrés au tableau 2.

Tableau 2 Matériaux/procédés de fabrication et poids des composants

Matériau/Procédé de fabrication	Poids des composants	
	kg	lb
Tambour en fonte	50,8	112
Disque en fonte	27,2	60
Tambour en composite de RNCan – infiltration dans une préforme	21,3	47
Disque en composite de RNCan – infiltration dans une préforme	11,8	26
Tambour en CMM – billette de CMM	21,3	47
Disque en CMM – billette de CMM	11,8	26
Tambour en CMM – CMM mélangé sur place	21,3	47
Disque en CMM – CMM mélangé sur place	11,8	26
Disque en matrice carbone renforcée de céramique de carbure de silicium	10	22

Pour chaque matériau/procédé évalué, un modèle de coûts a été élaboré àpartir des facteurs d'intrant, du coût des matériaux, des taux de récupération et de rejet, du coût et de la capacité des équipements, des besoins en main-d'œuvre et de la consommation d'énergie, entre autres. Règle générale, ces modèles obéissent aux grandes étapes de la fabrication. Le tableau 3 et les figures 5 et 6 présentent respectivement des exemples d'un schéma de fabrication, d'un état des coûts et d'une analyse de sensibilité. Il est ànoter que tous les coûts sont en dollars US.

Les coûts des matériaux ont été calculés en fonction des dimensions des pièces et des taux de récupération et de rejet àchaque étape du procédé. Pour établir les coûts totaux de main-d'œuvre, les chercheurs ont déterminé le nombre d'opérateurs àaffecter àchaque procédé/machine et le nombre de travailleurs indirects nécessaires au fonctionnement de l'usine. Ces coûts ont été répartis sur l'ensemble de la production. Il a été fait de même des dépenses d'immobilisation, des coûts d'énergie et des autres coûts associés àla fabrication du composant. Enfin, la sensibilité des coûts de fabrication aux facteurs d'intrant a été établie.

Tableau 3 État des coûts – Tambour en CMM fabriqué par moulage en coquille àpartir d'une billette

TAMBOUR DE FREIN CMM DE DURALCAN – CAMION DE CLASSES 7/8

Moulage en coquille de CMM en billette

		Coût par pièce ($US/pièce)	**Coût par année** ($US/année)	**Investissement**
Coûts variables				
	Matériaux	67,94 $	13 588 203 $	
	Main-d'œuvre directe	3,81 $	762 048 $	
	Énergie	2,28 $	455 768 $	
	Total – Coûts variables	74,03 $	14 806 019 $	
Coûts fixes				
	Équipement	1,32 $	264 333 $	3 965 000 $
	Outillage	0,24 $	48 040 $	
	Locaux	1,25 $	250 000 $	
	Main-d'œuvre indirecte	0,95 $	189 000 $	
	Total – Coûts fixes	3,76 $	751 373 $	
	GRAND TOTAL	**77,79 $**	**15 557 392 $**	

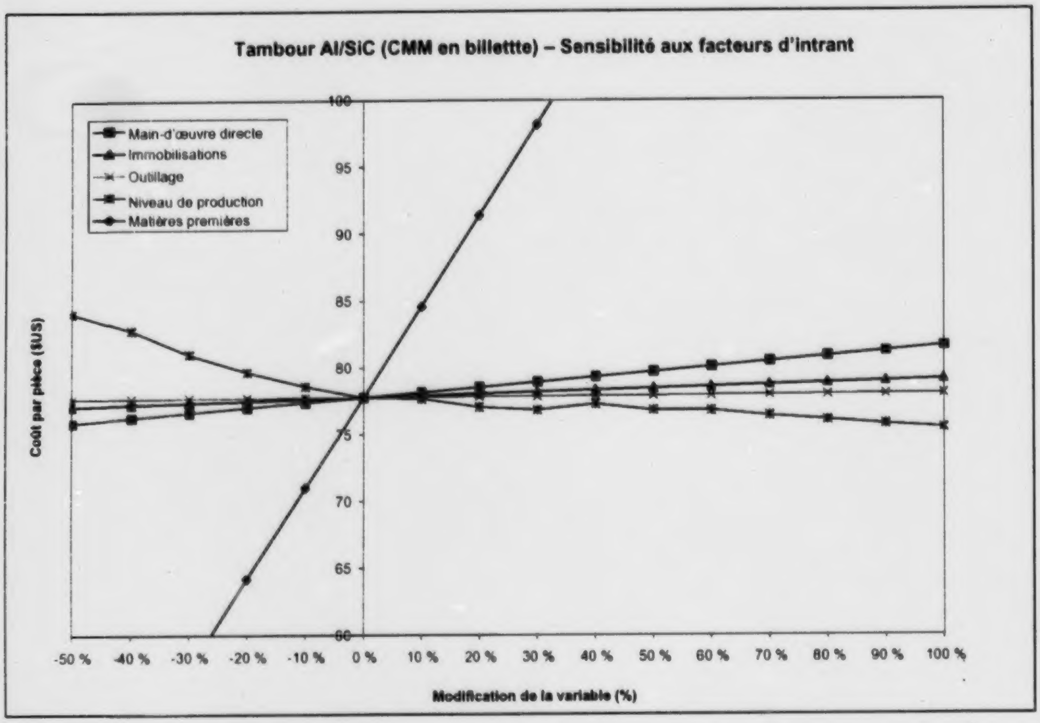

Figure 5 Tableau de sensibilité - Tambour MC fait à partir d'un CMM en billette

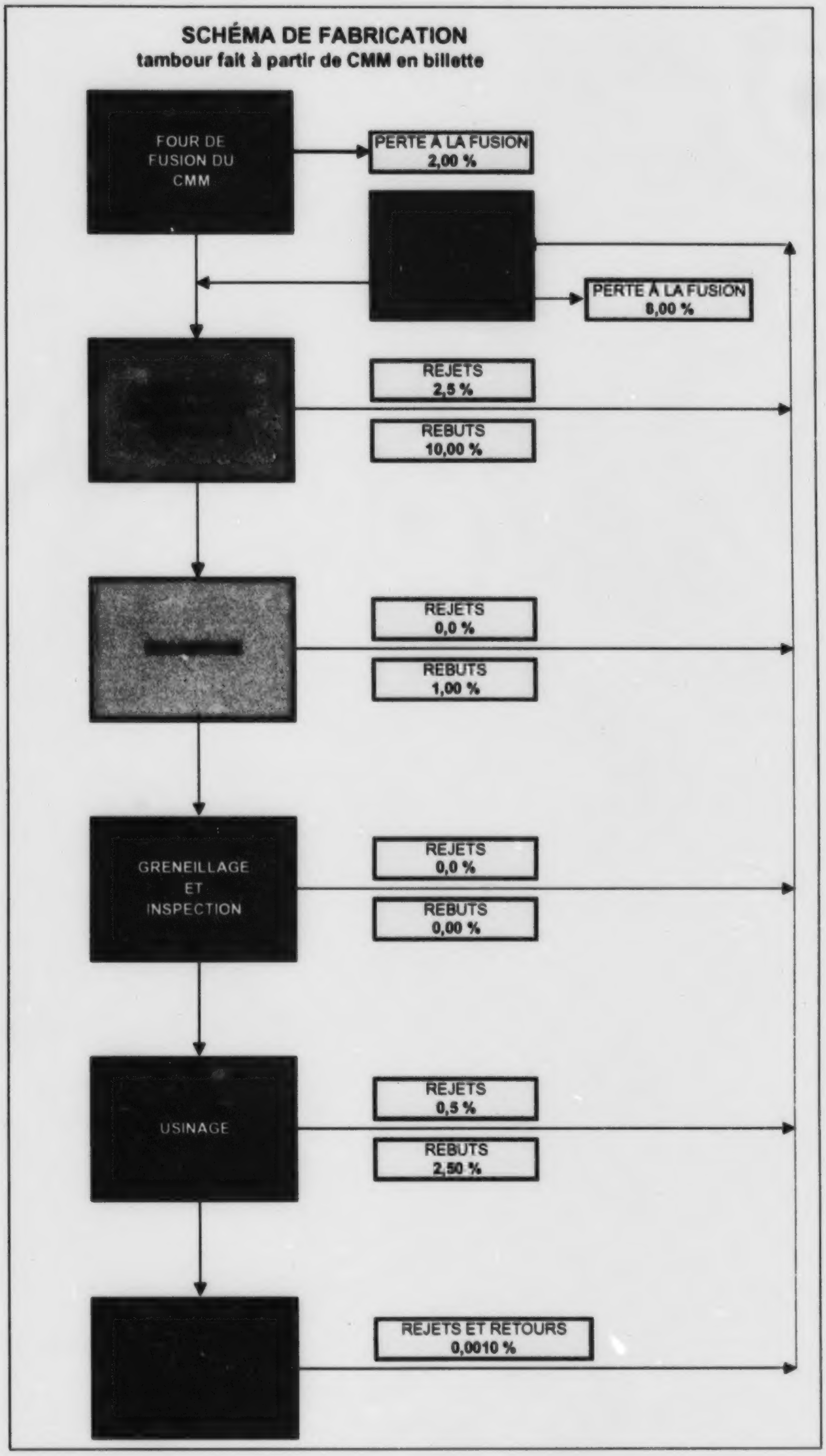

Figure 6 Schéma de fabrication – Moulage en coquille d'un tambour à partir de CMM en billette

Tableau 4 Matériaux/procédés de fabrication et coûts de fabrication

Matériau/procédé de fabrication	Coût de fabrication unitaire		Poids unitaire		$CAN/kg	Coût/allégement ($CAN/kg)	
	$US	$CAN	lb	kg		Tambour	Disque
Tambour en fonte	32,2 $	48,8 $	112	50,8	1,0 $		
Disque en fonte	31,8 $	48,2 $	60	27,2	1,8 $		
Tambour en composite de RNCan – infiltration dans une préforme	56,9 $	86,2 $	47	21,3	4,0 $	1,3 $	
Disque en composite de RNCan – infiltration dans une préforme	40,8 $	61,8 $	26	11,8	5,2 $		0,9 $
Tambour en CMM – billette de CMM	77,8 $	117,9 $	47	21,3	5,5 $	2,3 $	
Disque en CMM – billette de CMM	46,7 $	70,7 $	26	11,8	6,0 $		1,5 $
Tambour en CMM – CMM mélangé sur place	57,0 $	86,3 $	47	21,3	4,0 $	1,3 $	
Disque en CMM – CMM mélangé sur place	35,8 $	54,3 $	26	11,8	4,6 $		0,4 $
Disque en matrice de carbone renforcée de céramique de carbure de silicium	156,0 $	236,4	22	10,0	23,6 $		10,9 $

Le tableau 4 présente les résultats de l'analyse des coûts de fabrication des types choisis de tambours et de disques fabriqués à partir de divers matériaux et selon divers procédés. Dans tous les cas, c'est le coût des matières premières qui influe le plus sur le coût du produit final.

L'analyse révèle que tous les composants sont plus coûteux lorsqu'ils sont fabriqués en CMM plutôt qu'en fonte (que l'on considère le coût total ou le coût au kilogramme). Cette constatation est particulièrement claire dans le cas du tambour, dont la fabrication est plus simple mais exige davantage de matière première.

L'obstacle que pose le prix des CMM se trouve passablement tempéré si on calcule le coût par kilogramme d'allégement – une donnée prise en compte sur le marché (voir les deux dernières colonnes du tableau 4). Dans le cas des tambours, les procédés d'infiltration dans une préforme et de mélange sur place aboutissent aux coûts différentiels les plus bas, soit 1,3 $CAN/kg. Le coût de l'allégement est beaucoup plus élevé (2,3 $CAN/kg) dans le cas du CMM prémélangé (en billette). Pour ce qui est des disques, l'option de mélange sur place représenterait les coûts les plus faibles (0,4 $CAN/kg). Le procédé d'infiltration dans une préforme coûterait un peu plus que le double (0,9 $CAN/kg) du procédé de mélange sur place, tandis que le recours au CMM prémélangé coûterait 1,5 $CAN/kg. Finalement, le C/SiC est le matériau dont l'allégement coûterait le plus cher, soit 10,9 $CAN/kg.

Dans tous les cas, c'est le coût des matières premières qui s'est avéré avoir le plus d'effet sur les coûts unitaires. Le procédé d'infiltration dans une préforme réduit quelque peu les coûts d'usinage. Pour la fabrication des tambours, qui sont plus gros que les disques et représentent donc une plus grande consommation de matériau, ce procédé coûterait à peu près l'équivalent d'un procédé de mélange sur place de CMM/retraitement des rebuts. Le procédé de mélange sur place est moins coûteux dans le cas de la fabrication de disques.

Ces coûts sont des prix départ usine. Il faut les multiplier par un certain facteur pour avoir une idée du prix au détail. Plusieurs variables influent sur ce facteur, mais pour les pièces automobiles, on utilise généralement un facteur de l'ordre de 2 à 2,5. Ainsi, nombre des options de matériaux associées à la configuration tambour atteignent presque le coût différentiel cible de 3 $CAN/kg nécessaire à une bonne pénétration du marché.

Conclusions

L'aptitude des CMM à être utilisés dans les tambours et disques de freins pour camions peut être évaluée selon deux critères – leurs performances et leur prix. Le critère des performances a été évalué par une analyse des propriétés physiques et des performances des trois matériaux candidats - CMM, composite de CANMET et C/SiC. Ces critères doivent être tous les deux remplis pour que le produit soit viable.

L'analyse des propriétés physiques des matériaux a révélé que les composites à matrice d'aluminium conviendraient à la fabrication de tambours, mais non de disques, de freins pour camions. Il a en effet été déterminé que la température élevée atteinte par les freins à disques dépasse la température maximale d'utilisation des deux matériaux à base d'aluminium. Quant au C/SiC, il conviendrait tant aux freins à disques qu'aux freins à tambour.

L'analyse du marché a révélé que plus de 90 p. 100 des camions actuellement en service sont équipés de freins à tambour mais que les freins à disque se généralisent rapidement en Europe. Il est possible que cette tendance se propage à l'Amérique du Nord, mais cela n'ira vraisemblablement pas sans de fortes incitations du côté de la réglementation ou des prix. Les acheteurs sont très sensibles aux prix : ils sont disposés à payer un peu plus cher à condition de bénéficier d'un allégement du véhicule ou d'une diminution des coûts d'entretien. Selon les données de l'analyse, si le coût différentiel entre les freins classiques et les freins allégés était inférieur à 3 $CAN/kg, on pourrait s'attendre à une percée importante des pièces en CMM tant sur le marché des véhicules neufs que sur celui des pièces détachées. Au-delà d'un coût de 3 $, le marché est beaucoup moins réceptif, mais dans certains créneaux, des acheteurs seraient prêts à payer un coût différentiel plus élevé. On estime, par exemple, qu'environ 20 p. 100 du marché accepterait un coût différentiel aussi élevé que 10 $CAN/kg.

Les résultats de l'analyse des coûts de fabrication des modèles choisis de tambours et de disques (faits en divers matériaux et selon divers procédés) a mené aux conclusions suivantes :

- dans le cas de tambours, les coûts marginaux les plus faibles (1,3 $CAN/kg) sont obtenus avec les procédés d'infiltration dans une préforme ou de mélange sur place du CMM;
- l'option qui fait appel au CMM prémélangé est beaucoup plus coûteuse (2,3 $CAN/kg);
- dans le cas des disques, les coûts les plus faibles (0,4 $CAN/kg) correspondent à l'option de mélange sur place;
- le procédé d'infiltration dans une préforme coûterait légèrement plus que le double du procédé de mélange sur place, soit 0,9 $CAN/kg, tandis que le procédé faisant appel au CMM prémélangé coûterait 1,5 $CAN/kg;
- le C/SiC est le matériau le plus cher (4,6 $CAN/kg) pour la fabrication de disques par mélange sur place.

Il est donc permis de conclure que l'utilisation de CMM à matrice d'aluminium – soit en mélange homogène avec SiC ou en structure sandwich avec Al_2O_3 – a une grande probabilité d'être techniquement et économiquement viable sur le marché des freins à tambour.

Stratégie de développement

La poursuite du développement de tambours et de disques de freins pour poids lourds présuppose un grand nombre d'acquis et exige d'importantes ressources. La présente section expose ces exigences et propose une stratégie de développement en vue de la commercialisation de freins pour poids lourds faits des nouveaux matériaux étudiés.

Voici trois grands objectifs auxquels il faudra s'attaquer :

1. Fabrication et essai de nouveaux prototypes de tambours, ce qui permettra de valider les résultats des essais avec maquettes et de garantir la capacité des produits à supporter les fortes températures et contraintes générées par un poids lourd ou un autobus en freinage.
2. Établissement d'un partenariat stratégique avec un fabricant de tambours, qui pourra, si le produit réussit les essais techniques, commercialiser celui-ci de façon efficace et économique.
3. Confirmation des modèles financiers. Au terme de l'évaluation technique, analyse en profondeur des coûts de production de chaque option, afin de confirmer la justesse du modèle de coûts de production.

Essais de détermination des performances

Un programme doit être entrepris pour développer et mettre à l'essai des freins à tambour en CMM ou en composite de RNCan, et ainsi qualifier la faisabilité de tels freins. Les essais doivent valider, d'abord dans les conditions contrôlées d'un laboratoire, puis sur une piste d'essai, la fiabilité et la durabilité du tambour de frein en CMM.

Dans le cas du CMM, le matériau ainsi que les procédés et les installations de mise en forme existent déjà Mais pour ce qui est du composite de RNCan, il reste à formuler et mettre à l'essai le matériau final afin de valider les données estimatives utilisées aux fins de l'analyse. Celui-ci devra alors être mis en forme par RNCan à l'aide d'un moule spécial de forgeage-moulage. Ainsi, deux programmes d'évaluation sont décrits ci-après.

Le tambour en CMM peut être fabriqué à l'aide du matériau et selon les procédés de mise en forme et de finition déjà utilisés par Hayes Lemmerz et Volvo. Si cette option est retenue, le plus rapide serait de collaborer avec les entreprises et les fournisseurs qui ont participé aux travaux antérieurs. La démarche ci-après suppose une telle collaboration :

1. Conception du tambour
2. Mise en forme et parachèvement du tambour
3. Choix du matériau pour les plaquettes
4. Essai sur freinomètre à rouleaux
5. Évaluation sur piste d'essai
6. Évaluation globale

L'évaluation du composite de RNCan comporte des étapes supplémentaires. RNCan doit se charger des travaux, en raison du caractère confidentiel de son matériau et des procédés. Les tâches qui mènent à la création des prototypes de tambours sont énumérées ci-après. Une fois les prototypes obtenus, les méthodes d'essai figurant à la tâche 3 de l'étude du CMM seront appliquées.

1. Élaboration d'éprouvettes de composite
2. Évaluation des éprouvettes
3. Choix du type de tambour de frein et du procédé de fabrication
4. Fabrication d'un moule de forgeage-moulage

5. Fabrication de tambours de freins en composite par forgeage-moulage

6. Analyse des données et compte rendu des résultats

Coût et calendrier estimatifs du programme

Le coût du programme de mise en forme et d'essai des prototypes en CMM est évalué à 100 000 $CAN (voir la ventilation des coûts au tableau 5). En gros, la moitié de ce budget couvrira les coûts directs associés àla mise en forme, la finition et l'essai ou l'analyse des matériaux. L'autre moitié couvrira les coûts de gestion du projet et les frais de déplacement associés àla coordination, au suivi et àla production de rapports.

Le calendrier prévu du programme est représenté àla figure 7. On prévoit qu'il faudra compter trois semaines pour la mise en forme des tambours. Quant aux essais NSVAC/FMVSS 121[2], ils prennent normalement deux semaines. Les essais sur piste devraient nécessiter quatre autres semaines. En tout, six mois sont prévus pour l'ensemble du programme d'essais.

Le programme de développement de produits de RNCan, qui assurera la mise au point et la fabrication de six prototypes en composite de RNCan et de six prototypes en CMM, est évalué à 120 000 $CAN (voir le tableau 6). Il s'étalera sur environ 12 mois. Une fois les prototypes livrés, il faudra prévoir encore 88 000 $CAN (et cinq mois) pour les essais en laboratoire et sur piste (100 000 $CAN moins les coûts directs de la mise en forme du CMM – 12 000 $CAN), soit un total de 208 000 $CAN.

Tableau 5 Ventilation des coûts du programme d'essai de tambours en CMM

				Coûts par tâche ($CAN)			
Titre	Tâche	Chef de projet	Total main-d'œuvre	Déplacements			Projet
	Honoraires journaliers (8 h/jr)	1 120 $			Divers*	Total partiel	Total
1	Plan d'essai final, participants, fournisseurs	6 720 $	6 720 $	4 480 $		4 480 $	11 200 $
2	Mise en forme des tambours (12)	0 $	0 $	0 $	10 800 $	10 800 $	10 800 $
3	Usinage et finition des tambours	0 $	0 $	0 $	1 500 $	1 500 $	1 500 $
4	Choix du matériau de frottement	4 480 $	4 480 $	2 240 $	2 000 $	4 240 $	8 720 $
5	Essais de freinage en laboratoire	0 $	0 $	0 $	7 200 $	7 200 $	7 200 $
6	Analyse des matériaux	3 360 $	3 360 $	0 $	2 000 $	2 000 $	5 360 $
7	Essais sur piste – NSVAC/FMVSS 121	5 600 $	5 600 $	2 600 $	12 000 $	14 600 $	20 200 $
8	Essais sur piste approfondis	3 360 $	3 360 $	2 360 $	15 000 $	17 360 $	20 720 $
9	Évaluation des matériaux et des performances	3 360 $	3 360 $	0 $	3 000 $	3 000 $	6 360 $
10	Production d'un rapport	4 480 $	4 480 $	0 $	500 $	500 $	4 980 $
	Total	**31 360 $**	**31 360 $**	**11 680 $**	**54 000 $**	**65 680 $**	**97 040 $**

* Divers : communications, messagerie, imprimerie et autres coûts directs

10 %	Imprévus	3 136 $
	Total	100 176 $

---

[2] La norme NSVAC/FMVSS 121, équivalente aux États-Unis et au Canada, établit les exigences relatives à la conception et aux performances des systèmes de freinage pneumatique pour les véhicules moteurs. En vertu de cette réglementation, un véhicule doit réussir un essai de freinage dynamique qui consiste à effectuer 10 arrêts consécutifs, c.-à-d. passer de 80,5 km/h à 24,2 km/h à un taux de décélération de 2,72 m/s², puis de 32,2 km/h à 0 km/h à un taux de décélération de 4,27 m/s².

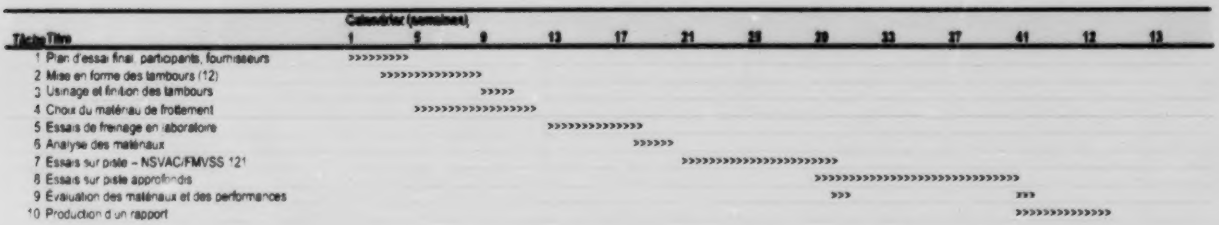

Figure 7 Calendrier du programme d'essai

Tableau 6 Ventilation des coûts de fabrication de tambours de freins en composite de RNCan

Principale tâche	Total partiel ($CAN)
Élaboration d'éprouvettes de composite	10 000 $
Évaluation des éprouvettes	15 000 $
Choix du type de tambour de frein et du procédé de fabrication	2 000 $
Fabrication du moule de forgeage-moulage	30 000 $
Fabrication des tambours de freins en composite par forgeage-moulage	45 500 $
Analyse des données et compte rendu des résultats	5 000 $
Matériaux et fournitures	8 000 $
Déplacements	3 500 $
Total	**120 000 $**

Partenaires possibles pour le développement du produit

Pour réussir la mise au point et le lancement d'un produit sur le marché des freins pour poids lourds, il importe de s'associer avec certains acteurs clés du secteur.

Fabricants/distributeurs

Comme il a été mentionné dans l'analyse du marché, le secteur des tambours de freins est occupé par une poignée d'équipementiers qui alimentent les constructeurs de véhicules neufs et le marché des pièces détachées en tambours seuls ou en tambours intégrés à des systèmes de freinage. Un des principaux rôles assignés à l'équipementier est d'assumer la responsabilité de la conception des produits. Il s'agit d'une responsabilité importante, qui rend difficile, voire impossible l'accès du marché à des entreprises naissantes. Sans compter que les équipementiers ont établi et contrôlent un réseau de distribution, de revendeurs, etc. Làencore, il faut beaucoup de temps et d'argent pour créer ce réseau de commercialisation et de distribution, d'où la nécessité, pour développer un nouveau produit quel qu'il soit de s'associer àun grand équipementier de façon que le nouveau produit bénéficie de la plus grande crédibilité technique et du marché le plus réceptif possible, lorsque sera venu le temps de le commercialiser.

Au cours de l'étude, l'équipe de projet a rencontré ou interviewé la plupart des équipementiers qui alimentent présentement le marché nord-américain. C'est Hayes Lemmerz qui possède l'expérience la plus récente des CMM. Même si l'entreprise a «mis sur la glace» le développement de ses produits, elle collaborerait volontiers à un tambour de frein de conception nouvelle, fabriqué à partir d'un nouveau matériau. Pour l'instant, elle consent à participer au projet d'évaluation. Sa contribution pourrait prendre la forme de modèles de moules existants, de temps d'essai en laboratoire et de soutien technique.

Fournisseurs de matériaux pour la mise en forme

Il existe plusieurs fournisseurs potentiels de CMM, dont Alcan, Norsk Hydro et MC21. À l'étape de la validation de principe, la provenance du matériau (Duralcan ou MC-21, les deux grands producteurs d'Amérique du Nord) importe relativement peu. Selon l'expérience de Volvo, il serait matériellement possible et peut-être même préférable d'utiliser un spectre plus étendu de dimensions de particules de SiC que celui qui entre normalement dans le produit Duralcan. Un spectre plus étendu a l'avantage de réduire les coûts de production (ce qui n'a pas d'importance à l'étape de l'évaluation) et d'améliorer la résistance de la matrice et le transfert thermique. Il a toutefois un inconvénient : plus les particules sont grosses, plus il est difficile de bien les mélanger et de les garder en suspension pendant le moulage.

Les discussions préliminaires tenues avec Duralcan et MC-21 ont été positives. Les deux entreprises se sont dites intéressées à participer au programme d'évaluation.

Fonderies

Le critère du choix de la fonderie pour la coulée du CMM devrait être une connaissance de la coulée des CMM et des tambours de freins pour usage industriel. Dans le cas des tambours de Hayes Lemmerz, le travail de coulée a été fait par la fonderie Eck et, pour des raisons de commodité et de continuité, il y aurait lieu de confier toute nouvelle tâche de moulage à Eck. D'autres usines ontariennes et québécoises de moulage de l'aluminium possèdent les capacités nécessaires pour fabriquer les préséries puis passer à la production en série, le moment venu. Le choix du fournisseur pour la production en série n'est pas considéré comme une priorité à cette étape de l'évaluation du développement du produit.

Fournisseurs du matériau de frottement

La pièce antagoniste du couple de frottement est un facteur clé du succès d'un tambour en CMM. Élaborer, pour les plaquettes, un matériau garantissant une bonne couche de transfert, de bons coefficients de frottement et un faible taux d'usure n'est pas une tâche facile. Un certain nombre de fournisseurs de matériaux de frottement ont participé à des programmes de freins en CMM par le passé et tous seront vraisemblablement intéressés à participer à un programme de freins pour véhicules lourds. Aucun des fournisseurs de matériaux de frottement n'est établi au Canada.

Constructeurs de véhicules

Les constructeurs de véhicules sont très sensibilisés aux avantages de systèmes de freinage allégés; certains grands constructeurs de camions ont manifesté de l'intérêt à l'égard du produit en développement. La réponse la plus enthousiaste a été reçue de Oshkosh Trucks (Wisconsin), qui a dit vouloir participer à un essai, quel qu'il soit, de tambours de freins. Oshkosh envisage la possibilité immédiate d'incorporer des tambours en CMM à ses bétonnières et, peut-être, à ses véhicules militaires. Aucun autre constructeur (d'autobus ou de camion) ne s'est montré aussi positif à l'égard d'une utilisation éventuelle du produit et du projet d'évaluation.

Le véhicule d'essai initial importe assez peu, sauf qu'il doit représenter un camion à essieu tandem de classe 8. Il serait avantageux de faire des essais de simulation d'un autobus urbain, soit en désactivant le deuxième ensemble de freins arrière, soit en ayant accès à un autobus d'essai.

Utilisateurs finals

Il est trop tôt, à l'étape de l'essai et de l'évaluation du prototype, pour associer de près les utilisateurs finals. Mais il serait utile de mettre sur pied un groupe d'utilisateurs qui pourraient donner leur avis sur les critères d'essai et de performance en freinage. L'existence de ce groupe consultatif accélérerait également l'organisation d'une démonstration en grandeur réelle, en supposant que le produit se serait révélé satisfaisant lors des essais en laboratoire. Le groupe d'utilisateurs devrait comprendre des conducteurs de camions qui travaillent dans des conditions exigeantes (bétonnières, camions-benne, etc.) et des conducteurs d'autobus.

Teneur en éléments canadiens

Le secteur de la fabrication de freins est dominé par une poignée de multinationales, dont aucune n'a son siège au Canada. Il existe bien des entreprises canadiennes de mise en forme, qui fabriquent des composants de freins, mais la plupart alimentent le marché des véhicules légers. Il n'existe, à l'heure actuelle, aucun fournisseur connu de tambours (ou disques) de freins pour poids lourds au Canada. On compte un certain nombre d'usines de transformation de l'aluminium, qui possèdent les capacités de mettre en forme le CMM (ou le composite de RNCan) et ces usines pourraient devenir des fournisseurs de pièces de freins dans l'éventualité où le produit objet de la présente étude serait commercialisé.

La plupart des ressources et des partenaires de la phase d'essai du prototype sont étrangers. Mais des fournisseurs canadiens pourraient être chargés de certaines tâches, comme la gestion du projet et les essais sur piste. De plus, il est probable que le gouvernement du Canada appuiera financièrement la phase ultérieure d'essai et de développement du produit, à condition que des contrats de fourniture de services de mise en forme ou de matériaux soient accordés à des entreprises canadiennes.

Recommandations

L'analyse effectuée dans le cadre de la présente étude a mené à conclure que l'utilisation de CMM dans des tambours de freins de véhicules lourds devrait être techniquement faisable et économiquement attrayante. Toutefois, les performances physiques de tels composants n'ont pas encore été démontrées et il est donc recommandé d'entreprendre à cette fin un programme de conception, de fabrication et d'essai de prototypes.

Les essais de performance des prototypes de freins fabriqués à l'aide des deux matériaux candidats (CMM et composite de RNCan) exigent des investissements différents de ressources. L'analyse des coûts et des calendriers d'essai des prototypes indique qu'il serait moins coûteux et moins long d'utiliser le CMM plutôt que le composite de RNCan pour fabriquer le produit, et que le développement serait moins incertain.

Il est recommandé ce qui suit :

- Qu'une entente de partenariat soit établie avec un fabricant de tambours de freins pour entreprendre le programme d'essai et d'évaluation de tambours de freins en CMM présenté ci-dessus. Selon les résultats de discussions préliminaires, il serait possible d'obtenir un soutien financier appréciable de la part de fabricants de tambours de freins et de fournisseurs de matériaux de frottement.

- Compte tenu des avantages marqués du composite de RNCan sur les plans technique et économique, il y aurait lieu, moyennant l'obtention de fonds supplémentaires, de soutenir également ce matériau, parce qu'il se prête à un procédé de fabrication moins coûteux et qu'il pourrait avoir une plus grande teneur en éléments canadiens, en raison des droits de propriété intellectuelle que détient RNCan à l'égard du procédé.

- Lorsque les essais sur prototypes se seront avérés concluants, il y aurait lieu d'organiser une démonstration du produit sur une plus grande échelle et de planifier de façon plus détaillée la production, après une analyse en profondeur des coûts de fabrication.

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GLOSSARY

ABS	Antilock braking system
ADI	Austempered ductile iron
Al_2O_3	Alumina
ASA	Automatic slack adjuster
ATA	American Trucking Association
CAMM	Centre for Automotive Materials and Manufacturing
CANMET	**Can**ada Centre for **M**ineral and **E**nergy **T**echnology
CCC	Carbon carbon composite
CCMTA	Canadian Council of Motor Transport Administrators
CLiMR	**C**anadian **L**ightweight **M**aterials **R**esearch **I**nitiative
CMVSS	Canadian Motor Vehicle Safety Standard
CSTT	Centre for Surface Transportation Technology
CTE	Coefficient of thermal expansion
C/SiC	Carbon/silicon carbide
DOT	Department of Transportation (US)
E	Young's modulus
EU	European Union
FEM	Finite element methods
FMVSS	Federal Motor Vehicle Safety Standard (US)
GCI	Grey cast iron
GVW	Gross vehicle weight
HPDC	High-pressure die casting
ICE	Inter-City Express
K	Thermal shock parameter (Eichelberg factor) Alternatively, kelvins when used as a unit
LPO	Landings per overhaul
MMC	Metal matrix composite
MOT	Maximum operating temperature
MTL	Materials Technology Laboratory
NRC	National Research Council Canada
NRCan	Natural Resources Canada
NVH	Noise, vibration and harshness
OE	Original equipment
OECD	Organisation for Economic Co-operation and Development
OEM	Original equipment manufacturer

ppm	Parts per million
RTV	Rotor thickness variation
S	Tensile strength
SiC/Al	Silicon carbide/aluminum
UK	United Kingdom
US	United States
WD	Warehouse distributor
α	Coefficient of linear expansion
δT	Temperature difference
σ	Thermal conductivity

1. INTRODUCTION

1.1 Background

This study was commissioned by the Transportation Development Centre of Transport Canada and is aimed at providing a broad technical review and competitive analysis of the use of lightweight metal matrix composite (MMC) materials in heavy-duty vehicle braking drums and rotors. As brake systems are typically a high-cost item to operators, especially for urban transit operators, the potential benefits of advanced materials in reducing wear, weight and maintenance incidence could be highly advantageous in decreasing costs, improving safety and decreasing energy consumption.

The materials that are specifically of interest in this study are MMCs of aluminum. There are two forms of this composite material:

- one that has been developed as a premixed homogeneous composite of aluminum and silicon carbide (SiC/Al) by Alcan Inc. under the trade name of Duralcan[1];
- a second non-homogeneous composite in which preformed inserts of a secondary material, usually alumina (Al_2O_3), are infiltrated with aluminum.

In both systems the goal is to introduce a strengthening and heat transfer medium into the aluminum so that it can be used in a greater range of stress and heat applications.

The use of lightweight materials in vehicles has been increasing as the needs for high fuel efficiencies and higher vehicle performance increase, and as the fabrication and application costs of alternate materials decrease. The amount of lightweight materials used in transport vehicles is directly proportional to the value the manufacturer or end user places on weight. Therefore, airplanes are largely made of lightweight materials such as aluminum, titanium, plastics, etc. Road vehicles have a lower content of more expensive lightweight materials, again depending on their application and market. A short summary of the current applications of MMC and carbon carbon composite (CCC) on transport vehicles is provided in sections 1.1.1 through 1.1.4.

1.1.1 Aircraft Brakes

Aircraft designers, constructors and operators constantly strive to reduce the structural weight of aircraft within economic limits. High-performance, vertical take-off and supersonic aircraft are all highly sensitive to structural weight. The lightweight and thermal characteristics of carbon-carbon composite materials make them ideally suited for use in aircraft braking systems. Aircraft brakes are of the full-face-contact type and use a number of rotor and stator discs to make up the disc stack. Within the brake the CCC material is used as a structural, thermal and frictional component (Figure 1-1).

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[1] Other aluminum producers, such as Norse Hydro, have formulated similar MMC materials

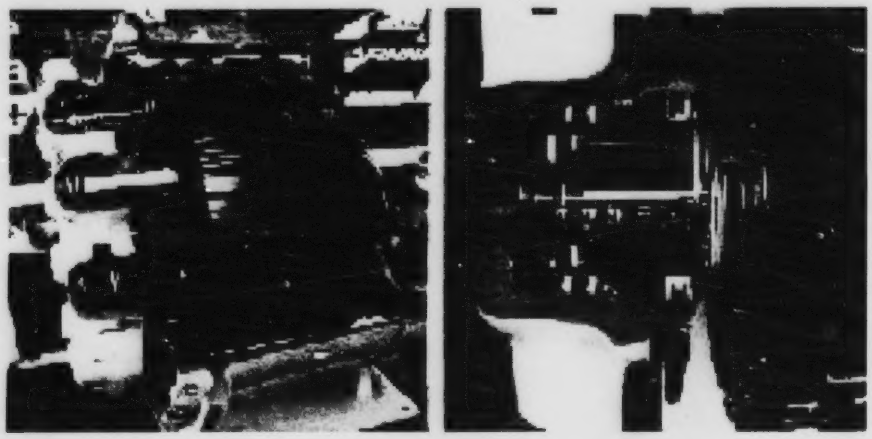

Figure 1-1 Typical Large Aircraft CCC Brake

Figure 1-1 depicts a cutaway aircraft brake mechanism showing the stacked rotor and stator discs (left) and the hydraulic piston assembly (right). Military aircraft such as the F-16, Mirage 2000, and Eurofighter Typhoon, as well as civil aircraft manufactured by Boeing, Airbus and Gulfstream, all exploit the advantages of CCC brake materials. The Concorde was the first civil aircraft to be fitted with CCC brakes as standard and now aircraft brake systems account for approximately 60 percent of the world's production of CCCs [1].

The weight saving available to aircraft constructors through the use of CCC brake discs allows savings to be passed on to those operating the aircraft, despite the high initial cost. The reduced weight of the aircraft increases the efficiency of aircraft operation and allows more fuel to be carried, resulting in a greater flight range. Alternatively, a greater payload may be carried, such as weaponry in the case of military aircraft, and passengers or freight in the case of civil aircraft. Reduced weight also affects performance, allowing faster rates of acceleration and more agile handling.

Although CCC brake systems offer large weight reductions as compared to metal-based friction pairs, there are a number of problems with CCCs:

- The lower tensile strength, impact resistance, strain to failure and thermal conductivity all present challenges;
- To compensate for the reduced density of the material and maintain sufficient thermal mass, the volume of the assembly needs to be up to 15 percent higher;
- There are significant changes in frictional coefficients between cold (very low coefficients) and high temperatures;
- Material costs are very high.

Although the component cost of CCC brakes is higher than that for brakes made from traditional brake materials, the cost of the material is reducing as the technology matures. When CCC brakes were first introduced, their cost was estimated at around C$1500/kg, whereas more recent estimates place the cost at about C$250 to $400/kg.

Despite the relatively high component price for CCC brakes, the use of these materials can be justified economically because of the reduced in-service costs, usually quantified as landings per overhaul

(LPO). The lower wear rate of the material during landing operations allows for the possibility of reduced aircraft maintenance time, which is an economic and operational advantage to both military and civil aircraft. A typical wide-bodied aircraft such as the Airbus A320 may expect to make approximately 1500 LPO using conventional metal brake materials, whereas using CCCs may allow 2500 to 3000 LPO [2]. The economics of aircraft are such that CCC has been gaining in market share over the last decade.

1.1.2 Motor-Sport Applications

The close relationship between motor-sport racing teams and production vehicle manufacturers has led to a number of technologies transferring from the race track to the road. In motor-sport applications, warm-up laps provide the opportunity to heat the CCC friction materials to their operating temperatures and thus avoid the poor friction performance associated with CCCs at low temperatures. Civil use of motor cars does not present this opportunity and for safety purposes a road vehicle's brakes are required to operate effectively and consistently in all weather conditions and despite recent braking history. Thus, the low coefficient of friction associated with CCCs at low temperature is a significant impediment to their use in production motor cars.

The governing bodies of motor sports determine the extent to which new technology can be used in the various sports to maintain the desired level of safety and competitiveness between the teams. These rules vary with time to reflect the cost of the technologies involved and to encourage continual development of the sport. The implementation of MMC and CCC friction materials is relatively expensive for a race team when compared to traditional materials. For this reason, only the sports with large budgets and the greatest demands for performance fully exploit these materials. Thus, only teams competing in Formula 1, LeMans and Grand Prix motorcycle racing are permitted the benefit of CCC clutches and brakes.

CCC brake discs and pads were introduced to Formula 1 by the Brabham team in 1982. These materials are now used universally within the sport. The present levels of vehicle performance could not be attained without the use of CCC brake materials. The increasing engine and tire performance of Formula 1 cars, together with improved aerodynamics and higher speeds, places increasing demands on the braking systems.

Motor-sport brake discs are vented and their operation is highly dependent on the flow of cooling air supplied to the brake system. Ducting is situated in an area of maximum airflow, which is fed to the centre of the disc and pumped radially outward through the vanes. This cooling is critical in preventing disc and pad material temperatures from exceeding the desired operating envelope (approximately 400 to 600°C) and thus leading to excessive oxidation [2].

Some application of MMC has occurred in the "near" motor-sport vehicles such as the Lotus Elise, where the weight advantage and the marketing appeal can support the increased costs. Also, both Daimler-Chrysler and Porsche have announced the introduction of Carbon/Silicon carbon brakes on their highest power models for the 2002 model year.

1.1.3 Rail Applications

SiC/Al MMCs have been used for components in high-speed train applications for some time. However, at approximately C$20/kg (for molten metal mixed composites), it is currently most cost-effective to concentrate on the replacement of the friction discs rather than the calipers and surrounding structure. SiC/Al MMC rail discs have a mass of approximately 70 kg compared to around 135 kg for traditional cast iron discs [3,4]. This can amount to a weight saving of approximately 13 tonnes per train when all discs are replaced. Figure 1-2 shows an example of an MMC railway disc brake.

As high-speed trains become increasingly important and their operating speeds increase, the need to reduce component weight will become a foremost concern. Following dynamometer testing by SAB WABCO in Germany, eight MMC disc brakes were run on the German Inter-City Express (ICE) for over 12 million km. Since 1997, an ICE with a full set of MMC discs has completed over 600,000 km.

The measured wear rate for rail MMC discs, over distances of 1.2 million km, has led to predictions that the discs should provide a life in excess of 15 years service. Therefore, the rail operator benefits not only from improved performance and energy efficiency, but also from reductions in vehicle maintenance time. Environmental controls on the operation of rail vehicles are becoming increasingly stringent. The use of SiC/Al MMC discs not only improves the vehicle's energy efficiency but also reduces the level of brake squeal associated with iron discs and the brake pad odour that is produced during high-energy stops. In addition to MMC-based designs, the French TGV Atlantique and the Germany ICE high-speed passenger trains have undergone tests with CCC brake systems. These materials can accommodate the higher energy loadings but come with significantly higher costs.

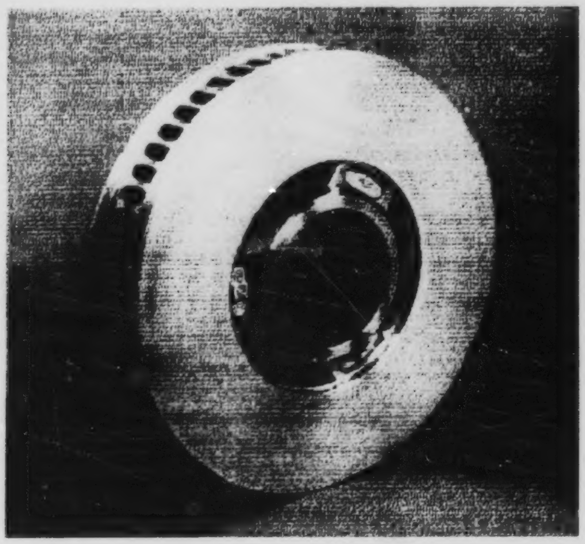

Figure 1-2 Example of an MMC Railway Disc Brake

1.1.4 Automotive and Truck Applications

In the area of medium and heavy-duty vehicle wheel and braking systems, lighter weight materials and designs have been introduced in a number of components including:

- aluminum wheels,
- aluminum wheel hubs, and
- steel/cast iron composite brake drums.

The market is attracted to these materials because they can result in decreased unladen weight, increased payload and, because of lower unsprung mass, better ride quality. These materials are also generally more expensive and thus have penetrated only portions of the heavy truck market.

As well, improved thermal performance of the brakes can be attained using lightweight materials such as aluminum MMCs, which have high specific heat capacity, or Carbon/Silicon Carbide (C/SiC) ceramic, which has high temperature capabilities. As yet, there are a limited number of applications of MMC or ceramic-based components in light-duty vehicles. These initial applications have been concentrated in high-end or speciality vehicles such as General Motors' EV1, Lotus's Elise, Daimler-Chrysler's Prowler and Volkwagen's Lupo for aluminum MMC, and Porsche and Mercedes Benz for C/SiC.

The use of MMC has been of interest in the passenger car sector, not only for its light weight but also for its acoustic damping characteristics, which significantly decrease brake squeal noise. Interest in the use of lighter weight materials or materials that have high noise and vibration damping characteristics is still a high priority in the auto and truck industry, as exhibited by the continuous series of technical papers written on brake noise, vibration and harshness (NVH), and on weight reduction techniques.

The expansion of the use of MMC in automotive applications has been limited by two main factors:

1. The physical performance of the material, in which there have been numerous physical failures due to high wear rates caused by excessive temperatures. The high temperatures have been attributed to a number of design issues, including alloy characteristics, formation of friction layer, and component sizing [5].
2. Cost structures, which have placed the finished components at a higher net cost than the incumbent cast iron designs, thus restricting the use of the new materials to a very limited number of vehicles.

In the truck brake area, the product development experience has been somewhat analogous to that in the passenger car area. All of the heavy-duty brake manufacturing companies interviewed during this study[2] acknowledged that they had tested MMC drum products. All had decided not to proceed with further product development because of costs and material performance concerns. In one case, a prototype drum was cast and performed adequately in laboratory tests but failed catastrophically during the early stages of a test track test.[3] Analysis of the failure indicated that it was probably due to a design flaw in the casting mould, but it was enough of a problem that the company discontinued the development work.

During the study, Volvo Technical Development Corp. published information [6] on work that it had undertaken on changing the particle size distribution of the MMC and using surface etching to increase the concentration of SiC at the friction surface. These changes made significant improvements in the formation and stability, and may indicate a technical pathway to improved thermal stability, which may allow the use of the MMC material in disc brake systems.

1.2 Study Scope

This study was tasked to assemble and analyse the current body of knowledge and application experience of MMCs and their competitors. In addition, the study compiled a profile of the current truck and bus market as well as typical brake component cost pictures such that comparative analysis and commercial viability assessments of the MMC materials can be made.

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[2] Hayes Lemmerz, ArvinMeritor, Ford, General Motors, Daimler-Chrysler, Brembo.

[3] Comments by engineering staff at Hayes Lemmerz.

Based on the analysis, the study also developed a "business case" for further promotion and development of these technologies and, based on the findings, a series of product development options are recommended together with their costs and commercialization timeframes.

1.3 Study Team

The study was conducted with personnel from three organizations:

National Research Council Canada (NRC) / Centre for Surface Transportation Technology (CSTT)

CSTT is part of Canada's National Research Council – a federal government organization with a mandate to conduct leading-edge research and development in support of Canadian industry. CSTT is one of NRC's fully cost-recoverable Technology Centres, providing independent consulting engineering services to national and international clients in the highway and railway sectors of the transportation industry for nearly 40 years. It has a staff of more than 70, with a broad range of expertise and experience in both railway and highway vehicle systems engineering.

CSTT was responsible for overall project management; the compilation of market data, brake costs, operation environments and business case analyses; and development of the Phase 2 R&D program.

Natural Resources Canada (NRCan) / Materials Technology Laboratory (MTL)

MTL is one of five laboratories that form the Canada Centre for Mineral and Energy Technology (CANMET), a major research and technology arm of NRCan. MTL focuses on developing and deploying technologies to improve all aspects of producing and using value-added products from metals and metallic materials. Areas of expertise include advanced materials, casting, energy-efficient metal production, advanced concrete and infrastructure reliability. MTL has start-to-finish capability covering the complete materials cycle from materials selection, fabrication and characterization to prototype product fabrication.

Located in Ottawa, MTL is the largest research centre in Canada dedicated to metals and materials fabrication, processing and evaluation. A technical staff of almost 100 scientists and technicians, working in 6,000 square metres of well-equipped laboratories, is engaged in a research and development program designed to provide technology solutions for Canadian industry. MTL is also the national agency for the certification of personnel for non-destructive testing.

MTL is unique in having the capability to cover the complete materials cycle – from materials selection, fabrication, in-service performance and characterization, to prototype product production. MTL's facilities include an experimental casting laboratory, a pilot-scale rolling mill, formability testing equipment, corrosion testing, mechanical testing and an extensive array of characterization facilities. It is possible for MTL to investigate a wide variety of materials, including metals and alloys (steel, aluminum, magnesium, copper, nickel, zirconium, etc.), metal matrix composites, intermetallics, coatings, ceramics, concrete and some plastics. It can simulate manufacturing processes like casting, rolling, stamping, forging, drawing, and welding, as well as advanced manufacturing processes such as metal powder injection moulding.

On this study, MTL undertook the physical evaluation of the materials by completing the following tasks: compiling information on competitive materials, reviewing prior scientific art and patents, and analysing the physical applicability of the materials to heavy-duty brake drums and rotors.

Centre for Automotive Materials and Manufacturing (CAMM)

CAMM is a joint project of industry, Queen's University and the Ontario Government. CAMM opened in January 1999 with a mandate to advance the competitiveness of automotive parts manufacturers. CAMM's facilities comprise 50,000 square feet of laboratory and classroom space, and staff drawn from various supporting organizations plus some common staff.

CAMM's research program is developed by an industry-led Research Advisory Committee, which has representatives from 3M Canada, Alcan International, CLiMRI, Dofasco, DuPont Canada, Magna International, Maksteel, McMaster University, Queen's University and the University of Toronto.

For this study, CAMM compiled production cost data, developed manufacturing cost models for the competitive materials, and assessed the likely cost differentials and key cost variables. It also provided in-kind financial support for its work.

1.4 Report Structure

The remainder of this report is divided into four main chapters that correspond to the four primary work packages.

Chapter 2 reviews the technology and current market for heavy vehicle brake components in North America and Europe and provides an estimate of the possible economic value of the new brake materials' attributes (weight and wear).

Chapter 3 provides an analysis of the alternate materials and their physical capabilities to be used for heavy-duty truck brake drums and rotors.

Chapter 4 details an economic modelling of production costs for the alternate materials.

Chapter 5 presents an analysis of the technical and economic probability of a successful brake drum or rotor product and a development plan to achieve success.

2. HEAVY-DUTY BRAKE TECHNOLOGIES AND MARKETS

This chapter provides an overview of the current technologies for heavy-duty vehicle brake components, market size and competitive factors. It starts by detailing the brake components and the variety of brake systems in use. This is followed by a review of the cost structures of brake operation and maintenance that provide a cost framework on which to judge the new materials' competitiveness.

2.1 What Brakes Must Do

The basic role of vehicle brakes is to absorb and dissipate energy of a vehicle in motion. They do this by generating friction between the rotating wheel brake drum or rotor and a friction block that is forced against the drum or rotor. The energy that is absorbed is in the form of kinetic (i.e., change of speed) or potential (i.e., change of elevation) energy and this gives rise to different braking "mission profiles" as will be discussed later in the chapter.

Demands on braking systems have been increasing dramatically over the past 50 years. Data compiled by TRW Chassis Systems [7] and shown graphically in Figure 2-1 illustrate these changes for passenger cars (trucks are similar). The major conclusion of the TRW study was that there had been a 30 percent increase in the braking power needed as a result of:

- higher posted speeds,
- higher gross vehicle weight (GVW) limits,
- a 30 percent decrease in the vehicle's natural retardation versus 15 years ago,
- a 19 percent reduction in engine braking force, and
- a trend to "low-profile" vehicles.

During braking, the kinetic energy of the vehicle is converted into thermal energy at the friction surfaces (e.g., linings and drums) of the brake and at the tire-roadway interface. An ideal brake system is one that has a high capacity to convert kinetic energy (the rotating wheel) into heat. This is termed the thermal capacity of the brake. Moreover, the ideal brake has a high cooling rate whereby heat is convected rapidly away from the brake in preparation for the next brake application.

When brakes are hot, they are less able to convert additional kinetic energy into heat and thus lose their effectiveness. This is termed brake fade. Brake fade is not likely to occur when brake applications are brief and infrequent. However, when brake applications are frequent or continuous for a period of time (e.g., during mountain descents), brake fade becomes a significant consideration.

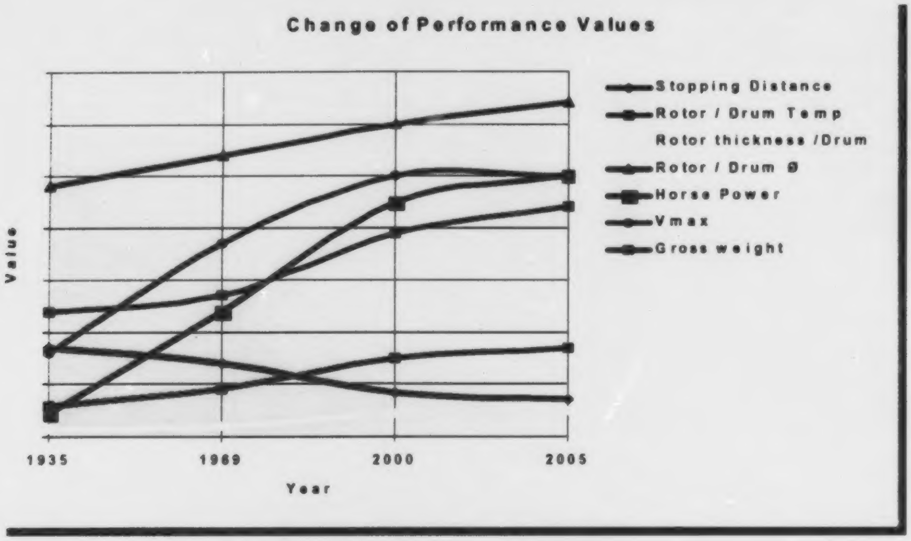

Figure 2-1 Evolution of Passenger Vehicle Brake Performance Demands

Brake fade is the immediate negative safety consequence of overheated brakes, but there is also a longer term consequence. Hot brakes wear much faster than cool brakes. It has been estimated [8] that a brake operating at 200°C wears four times faster than one operating at 93°C. Furthermore, a brake operating at 315°C wears nearly seven times faster than one operating at 93°C.

The large braking forces needed to stop a heavy truck generate a correspondingly high amount of heat. Truck brakes are obviously much larger in absolute size than passenger vehicle brakes, but in relation to vehicle weight they are smaller. In particular, the contact surface of the brake drum (the "swept area") is considerably smaller (about a half to a quarter of the size relative to vehicle weight) in air-braked combination units than in passenger cars, as seen in Table 2-1. Since truck brakes are smaller in relation to vehicle weight, they have correspondingly less thermal capacity than passenger vehicle brakes. Also, as the application forces are much higher, the specific power (i.e., power per swept area) that they have to absorb is in the order of 400 percent greater than that of a car.

Table 2-1 Comparison of Passenger Car and Truck Disc Brakes

Vehicle type	*Brake lining area cm^2/kg of GVW*	*Swept area cm^2/kg of GVW*	*Specific power kW/m^2*
Passenger Cars	0.40	1.21	67
Tractor Trailers	0.36	0.57	300
Ratio	0.9	0.5	4.5

Similar comparative data [9,10] of the relative lining area (Figure 2-2) and energy loading (Figure 2-3) between disc and drum brake systems show the inherent differences between the two brake configurations.

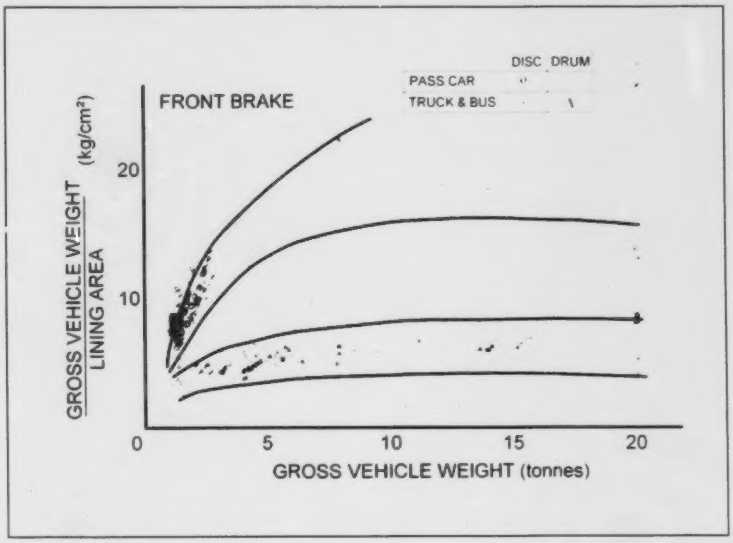

Figure 2-2 Specific Lining Area Comparison between Disc and Drum Brakes

This brake size imbalance between trucks and cars (disc and drum) has been exacerbated by the fuel economy-driven objective of achieving low parasitic drag. Low parasitic drag means that less engine power is needed to move the vehicle down the road. However, lowering parasitic drag has resulted in greater demands and stresses on the braking system.

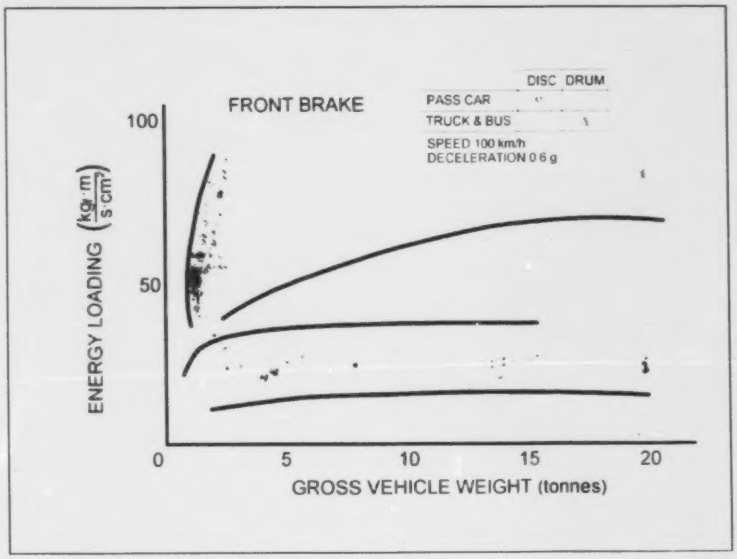

Figure 2-3 Energy Loading Comparisons between Disc and Drum Brake Systems

Take the example of a vehicle descending a grade. The vehicle will accelerate unless retarding force is provided. Retarding force can come from two sources: parasitic drag and braking. Parasitic drag is caused by rolling resistance, aerodynamic drag and engine drag. Depending on the slope of the grade and gear range used, parasitic drag by itself may be sufficient to control the speed of the vehicle. If, however, it is not sufficient, the brakes must be applied to supplement the parasitic drag and provide the necessary overall retarding force.

Table 2-2 shows parasitic drag as measured for three different types of vehicles at 90 km/h:

- a 1,364 kg passenger car,
- an 11,364 kg truck, and
- a 36,364 kg tractor-trailer.

In addition to drag force, Table 2-2 [11] shows the normalized drag force (i.e., the drag force per kilogram of vehicle weight). The tractor-semitrailer has nine times the drag of the car, but only one-third the drag in relation to vehicle weight. This means that the truck will roll much more easily relative to its own weight than a car.

Table 2-2 Vehicle Drag Comparison at 90 km/h

Vehicle Type	Gross Vehicle Weight (kg)	Drag Force (kg)	Drag/GVW ratio
Passenger Car	1364	75	0.055
Medium Straight Truck (Loaded)	11364	432	0.038
Tractor-Semitrailer (Loaded)	36364	636	0.018

A vehicle that rolls easily requires greater applied force to stop. Figure 2-4 shows the horsepower that must be absorbed/dissipated by the braking system of each of the three vehicles at 90 km/h (55 mph) normalized for vehicle weight, as a function of percent grade. For each vehicle type, no braking is required to maintain 90 km/h up to a certain threshold grade level. Beyond that threshold, linear increases in braking power (normalized for vehicle weight) are required with increases in percent grade.

Figure 2-4 [11] shows that the tractor-semitrailer grade threshold for the application of brakes is 1.8 percent, whereas the passenger car's threshold is 5.3 percent. For a steep 6 percent grade, the kW per 100 kg required for the three vehicle types are as follows:

- Passenger car (3000 lb. or 1361 kg): 0.09 hp/100 lb. or 0.03 kW/kg
- Medium truck (25,000 lb. or 11,340 kg): 0.32 hp/100 lb. or 0.11 kW/kg
- Tractor-semitrailer (80,000 lb. or 36,287 kg): 0.63 hp/100 lb. or 0.21 kW/kg

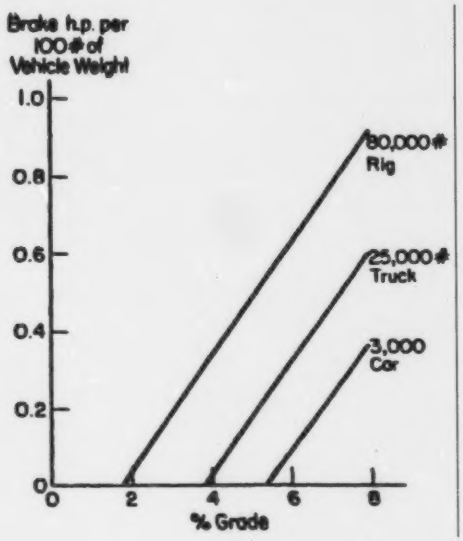

Figure 2-4 Braking Power to Maintain 90 km/h Speed at Percent Grade

The tractor-semitrailer weighs 26.7 times more than the car and, on a 6 percent grade, requires 7 times as much braking power per 100 kg of vehicle weight. Thus, the tractor-semitrailer requires almost 200 times as much braking power than the car under this downgrade descent condition.

Because brakes on trucks have lower thermal capacity (per kilogram) than passenger cars, trucks must be driven down steep grades at lower speeds to ensure that temperatures are maintained with operational bounds. When the brakes get hot, a number of things happen:

- In a drum brake, the drum "bells" out and this increases the distance that the pad must move for the same force. This in turn increases the "stroke" required from the air chamber which, because of its mechanical design, provides a decreasing amount of torque to the wheel, as seen in Figure 2-5 [12].

- The brake pad wear rate increases, as seen in Figure 2-6 [13]. The higher wear rates mean that the truck brake must be replaced more often and adjusted more frequently (either manually or by automatic slack adjusters). The increased frequency with which the brake stroke requires adjustment is one of the primary reasons for the high rate of safety inspection failures for brake stroke adjustment found in roadside surveys of trucks (i.e., with higher wear rates, the brakes are more likely to be out of adjustment unless a very rigorous maintenance routine is followed).

- This higher wear rate also decreases the effective friction coefficient (μ), which is partially responsible for the decrease in braking torque, as seen in Figure 2-5.

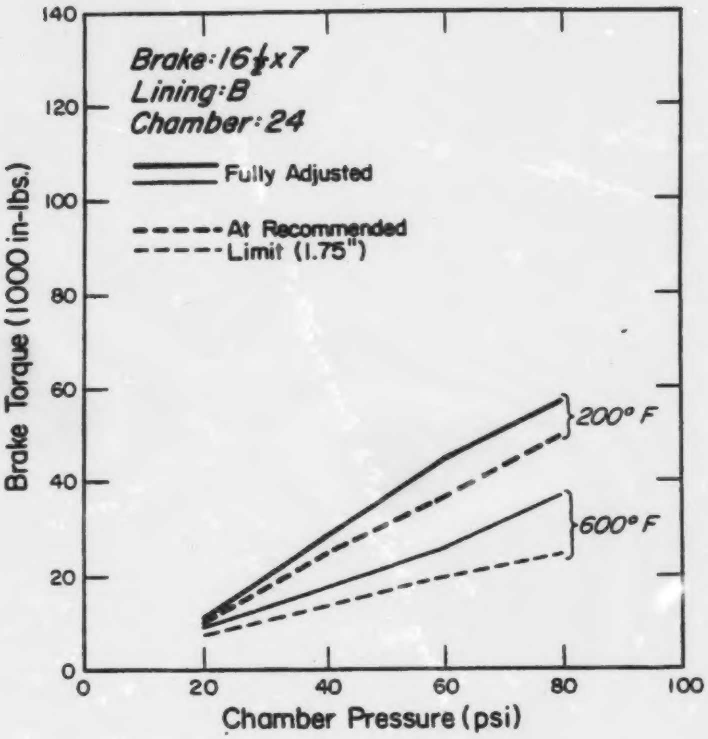

Figure 2-5 Brake Torque versus Pressure and Brake Temperature

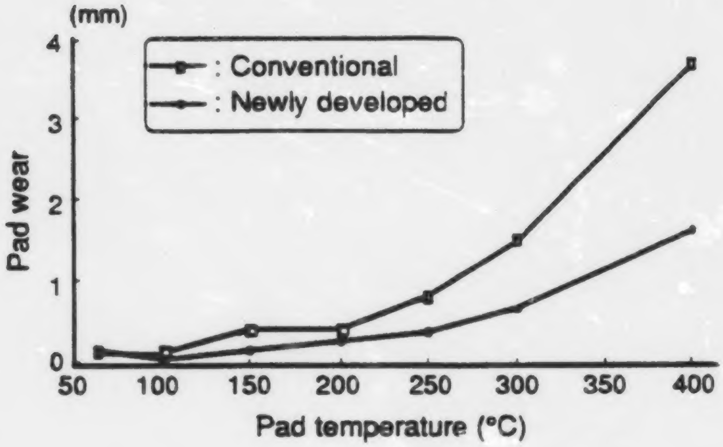

Figure 2-6 Brake Temperatures versus Pad Wear

While the extended "drag" braking condition is perhaps the most severe condition, most braking events are low-pressure, and thus low-force, events. This can be seen in Figure 2-7, which is based on a recent UK study of combined vehicle braking [14]. Over 90 percent of the brake applications are at pressures of less than 150 kPa in systems where the maximum application pressure is in the order of 600 kPa.

In summary, truck brakes have a much more severe duty cycle to satisfy than car brakes and have a relatively lower thermal mass through which to transfer the thermal energy. They are thus a much more demanding application for any material but also provide great opportunities for improved thermal and braking performance.

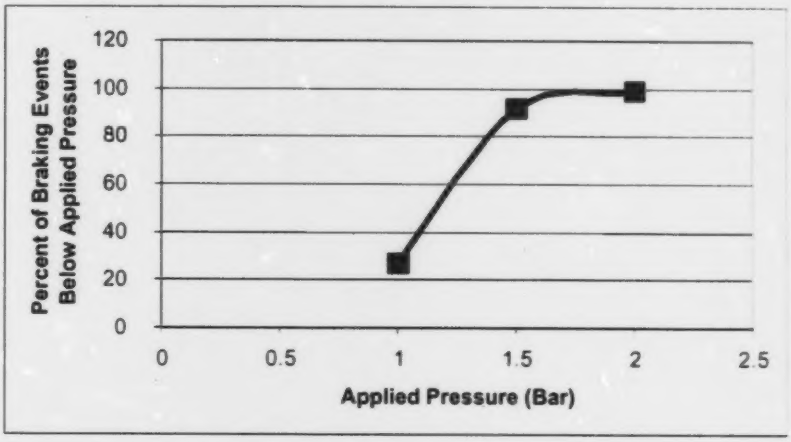

Figure 2-7 Distribution of Brake Application Force

2.2 Current Air Brake Technologies

This section provides an overview of the various configurations and components in current generations of heavy-duty brakes. It also provides information describing the physical performance of currently available alternate materials for other wheel-end components as an example of the benefits of material substitution.

Drum brakes were used on the front and rear wheels of most cars and trucks prior to the 1970s. They are still used extensively for the rear wheels of production cars and on commercial vehicles.

Drum brake mechanisms have a rotating drum attached directly to the wheel. Inside the drum are brake shoes that pivot around a stationary anchor point. The shoes are forced apart by a variety of methods, including:

- a rotating cam, typically S-shaped in the case of most truck systems and thus called an S-cam system (see Figure 2-8); or
- one or more pistons that prise two wedges or shafts apart, thus giving rise to the name wedge brakes (see Figure 2-9 and Figure 2-10).

Activation of the brake mechanism forces the shoes against the wall of the rotating drum. The brake shoes are lined with a friction material called the drum lining, which is composed of materials designed to give good friction characteristics when running against the drum rotor material. The anchor point is attached to a backing plate that is connected to the vehicle suspension. Therefore, during braking, the friction force generated at the brake shoes is transferred to the anchor point, which in turn transfers the braking load to the suspension.

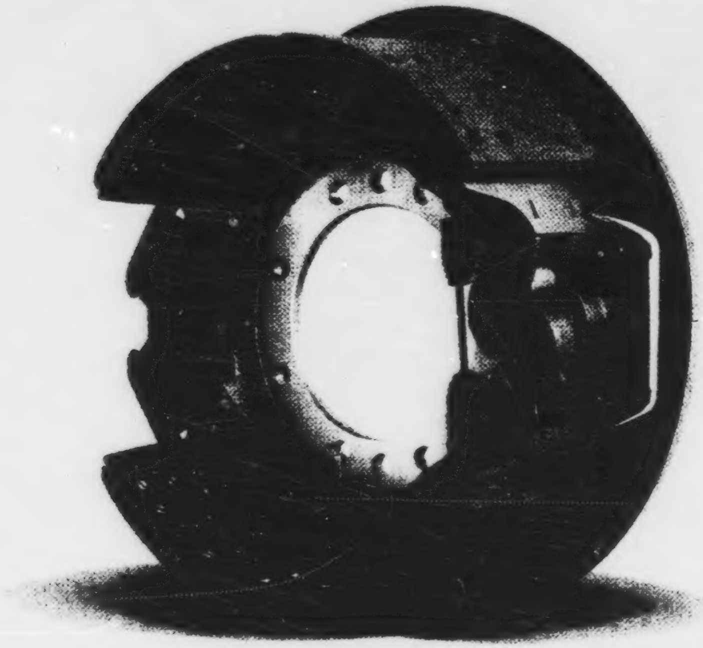

Figure 2-8 Typical Air S-Cam Drum Brake

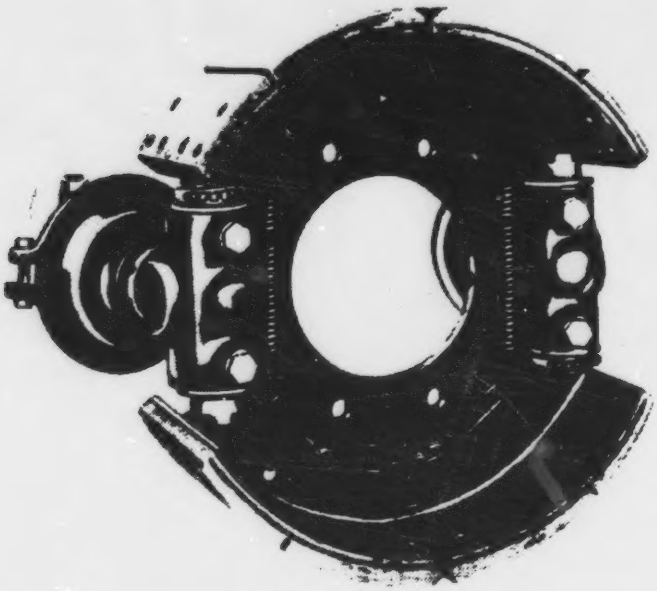

Figure 2-9 ArvinMeritor StopMaster Wedge Brake

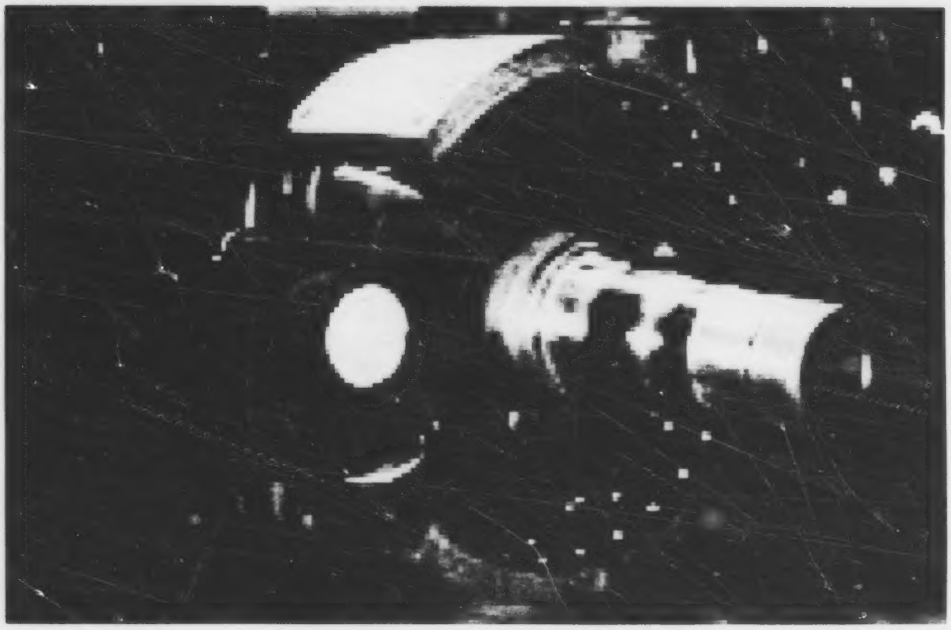

Figure 2-10 ArvinMeritor CamLaster Brake

Drum brakes are relatively less expensive to produce than disc brakes. However, they exhibit a number of potential problems:

- Drum brakes have a closed outboard face. Although this may offer some protection against the ingress of contaminants such as water or dust, once these contaminants are present on the friction surface of the drum, centripetal force holds them within the drum, on the friction surface.
- The diametrically opposed forces applied to the drum by the brake shoes may cause permanent deformation of the drum. This can lead to reduced contact of the friction lining with a reduction in braking performance.
- At extreme temperatures the drum design will "bell" (i.e., expand at the mouth), which introduces differentials in the applied forces and leads to a decrease in the average force and thus a decrease in braking power.
- As the sacrificial friction material of the pad wears, the brake must be adjusted to maintain the gap between drum and pad.[4] If the gap gets too large, the brake cannot develop the maximum braking force and thus maximum braking power. As such, brakes need to be periodically adjusted, either manually or through the use of automatic slack adjusters. Since drum brakes have pads only on one side of the drum, as opposed to two sides for discs, the rate of gap increase is double that of disc brakes; therefore, drum brakes must be adjusted approximately twice as frequently.
- Drum brakes have longer release times because of the one-sided nature of the design. This causes a marginal increase in the application times and a decrease in the time available for cooling.

The materials used for drums must exhibit a number of characteristics (these are described in more detail in Chapter 3), including:

- good friction properties,
- sufficient strength to withstand the braking application forces,
- high heat transfer and thermal capacity, and
- low cost.

The following text briefly describes the currently available material options for truck brake drums.

Cast Iron Brake Drum

The current industry standard brake drum is shown in Figure 2-11. It is fabricated using grey cast iron. The range of vehicle and axle sizes for trucks is limited in number by the variety of brake shoe drum configurations. The most common size is 419 mm x 178 mm (16.5 in. x 7 in.), although wider brake pads have been becoming more popular as a result of higher energy loads from the increased payloads (in Canada), speed and reduced aerodynamic drag. The drums are sold almost as a commodity, with all the North American original equipment (OE), and aftermarket suppliers purchasing their castings from third party casters, often from offshore low labour cost sources.

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[4] This gap is important for the proper operation of the brake and is connected to the overall brake system design.

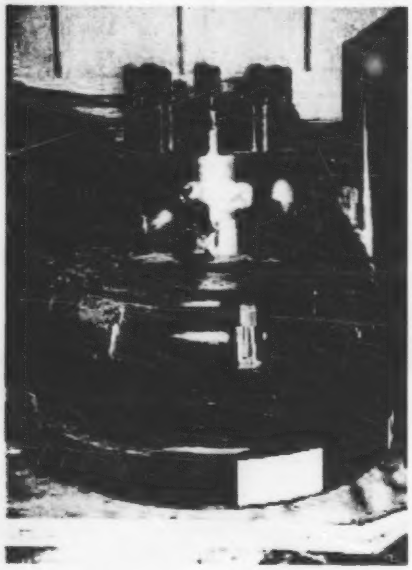

Figure 2-11 Typical Brake Drum (with hub)

Composite Steel/Iron Drums

A lighter weight drum alternative that has been on the market for over 20 years is the composite steel shell/cast iron drum. As can be seen in Figure 2-12, a steel containment rim is fabricated and then cast iron is centrifugally cast into the mould. This system results in weight savings of approximately 8 kg on a 50 kg drum. The design was originally developed for use on DC3 aircraft during World War II and has been manufactured by Hayes Lemmerz since then under the trade name of Centrifuge. In the last five years, ArvinMeritor has developed a competitor product with the trade name of Steelite.

The product market penetration is currently between 20 to 25 percent of the heavy truck market. Besides price premiums, one of the market impediments is the availability of the product in the aftermarket.[5] This leads to the replacement of these lightweight drums with standard drums as the vehicle gets older, apparently because of changing vehicle ownership and maintenance practices for these lower value vehicles.

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[5] Comment by Webb Wheel and Hayes Lemmerz personnel.

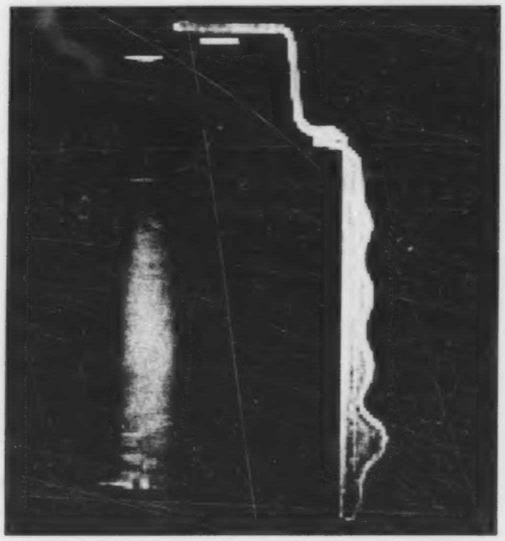


Figure 2-12 Cross Section of a Composite Steel/Iron Brake Drum

Ductile Iron Drums

A weight advantage can be achieved by casting the drum using austempered ductile iron (ADI) instead of standard grey iron [15]. This is a relatively new casting technique that has been used for other automotive parts and is now finding its way into brake components. ADI is a form of iron produced through a carefully controlled heat treatment of the iron casting, which produces structural and chemical changes in the iron. Because the material is stronger, a net weight advantage is achieved, which results in weight saving of approximately 10 kg on a 35 kg drum. Webb Wheel Corp., an Alabama-based company, has recently introduced this new type of drum [16]. The product is targeted at the same weight-sensitive users that make use of composite drums. The quoted retail price for the new Platinum drums is US$95, up from US$50 for Webb Wheel's standard cast iron product and a saved marginal cost of US$4.50/kg (~C$7/kg).

2.2.1 Caliper Disc Brakes

Caliper disc brakes offer considerably more stopping power than drum brakes and, consequently, are used on the front wheels of production cars, all four wheels of high-performance cars and the axles of high-speed railway bogies. Caliper disc brakes have a rotating disc attached directly to the wheel and static pads positioned on each side of the disc (see Figure 2-13). When the brake is activated, these pads are brought into contact with the rotating disc. The disc pads are composed of a friction material designed to give good frictional performance when running against the disc rotor. The brake caliper transfers the friction forces generated at the disc-pad interface to the vehicle suspension.

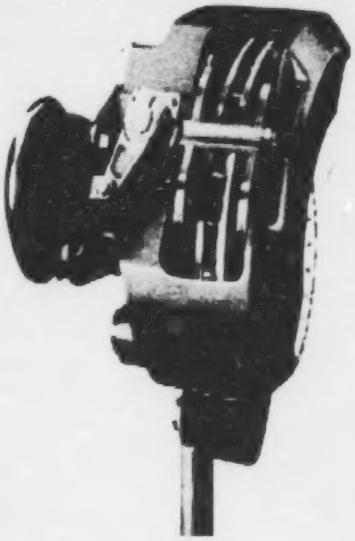


Figure 2-13 Typical Air Actuated Disc Brake

For light-duty vehicles the caliper is activated by hydraulic pistons. The number of pistons used varies depending upon the caliper design, although often ranging between one and three. In heavy-duty vehicles compressed air is used to activate the calipers. Most modern calipers are of the floating design, in which pistons are only mounted on one side of the caliper, generally the inboard side. Activating the piston(s) on the inboard side of the disc applies the inboard pad. The caliper then slides laterally until the outboard pad also comes into contact with the disc. Further activation of the pistons increases the compressive force on the disc, which is equal on both sides.

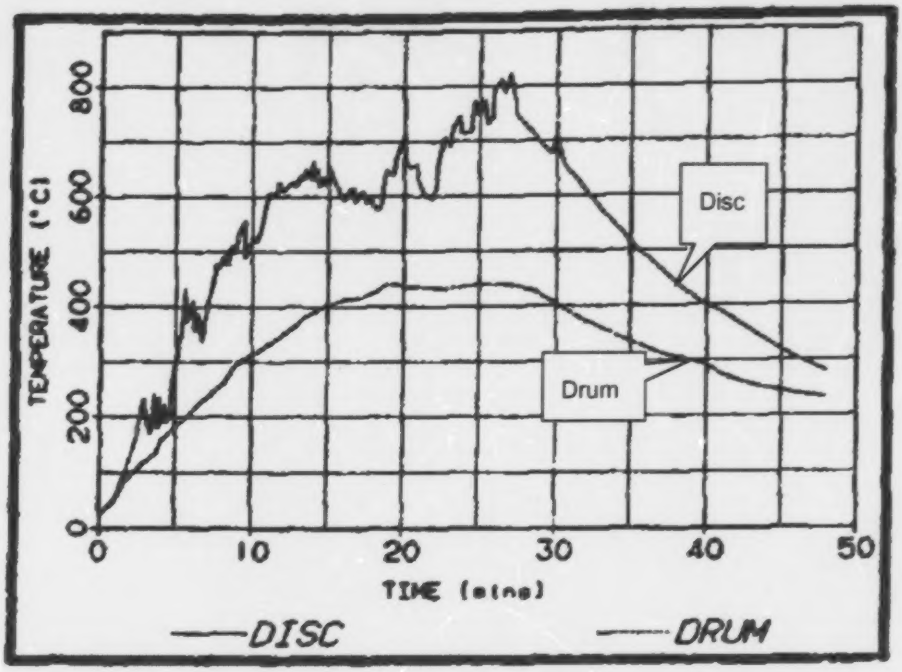
The symmetric pad forces on the disc prevent distortion of the disc, thus overcoming one of the limitations of drum brakes. Disc brake systems are less sensitive to contaminants because centrifugal force throws off water and dust as the disc rotates. One of the main advantages of the caliper disc brake over the drum brake is the cooling capability of the disc system. The increased area of the disc exposed to airflow, as compared to drum brakes, results in the disc running at a cooler temperature because of improved convection.

Variations to the basic disc design are usually made to improve the thermal behaviour of the components. The discs are commonly vented to allow cooling airflow to be pumped from the centre of the disc to the outside edge. In addition, the discs may be cross-drilled or grooved on the friction interface. This assists with pad cooling and weight reduction. Combinations of cross-drilling and venting are common on high-performance vehicles to prevent excessive disc and pad temperatures. In addition, discs may be floated. In this arrangement the disc is not constrained radially and a degree of thermal growth is permitted. This reduces the thermal stress in the disc and prevents disc distortion and associated problems.

The small disc pad area results in higher energy loadings and generally higher rotor temperatures than comparable drum systems [17], as shown in Figure 2-14. Even though the maximum temperatures are increased, the rate of cooling is also increased because of the larger surface area. Furthermore, the design allows for high application pressures, which decrease the fading effect of sustained high-energy operations.

The current air disc brake systems were originally introduced in the mid-1980s partially in response to the introduction of new stopping distance rules in the US. These requirements were eventually dropped and the US market for discs never developed. The experience of the operators at the time consisted of low durability, high wear rates and poor parts support.

Because of increased concern about heavy vehicle braking systems, the European Union (EU) stiffened its braking requirements in the 1990s. These regulations demand more sustained high-energy braking capacity (long downhill descents) and have resulted in high penetration rates of air disc brakes in the new vehicle market, with over 80 percent of new straight trucks and tractors and 50 percent of the trailers equipped with discs. This introduction has occurred despite a 10 to 25 kg weight disadvantage per wheel end, higher wear rates and higher costs (see section 2.3 for a fuller discussion of the European markets).



(Based on a 19T truck undertaking the standard European alpine (Rosseldstrasse) descent)

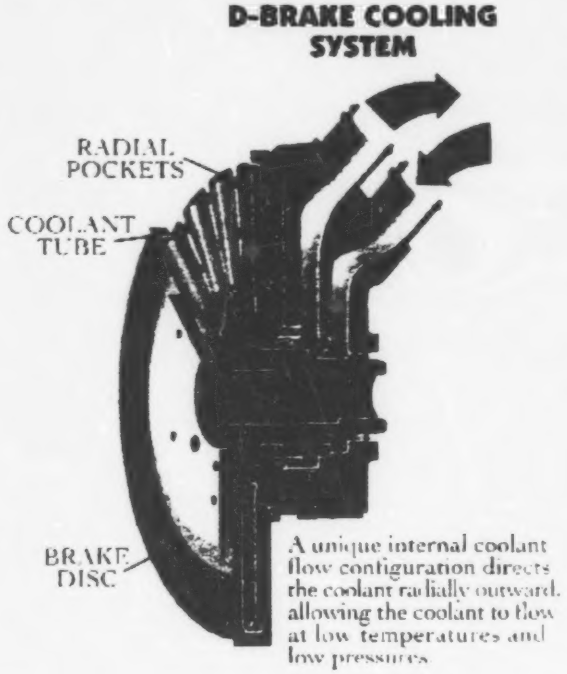
Figure 2-14 Temperature Comparisons of Drum and Disc Truck Brakes

2.2.2 Other Brake Technologies

There are a number of other brake technologies or designs that attempt to solve the biggest performance problem with truck brakes – heat. Unfortunately, all of these other alternatives are significantly more expensive than drum or disc brakes and have developed little or no market interest to date.

Active Cooling Systems

Another approach to decreasing heat and wear on brakes is to improve the heat transfer and heat rejection. This approach has been used by deConti Industries, a Connecticut-based company. DeConti's D-Brake system (Figure 2-15) uses coolant to absorb heat, carrying the resulting heat to a radiator and dissipating it to the ambient air. Thus, only the size of the cooling system and the flow rate of coolant circulated through the D-Brake system limit the heat dissipation capacity of the D-Brake. The system is engineered to ensure that the peak operating temperature of the disc surface does not exceed the design temperature limit, which is typically well within the capabilities of the friction materials. With lower temperatures, the friction surfaces last many times longer than those of conventional, air-cooled brakes, and improved braking performance and lower wear of the brake linings is achieved. Both in-driveline (supplemental) and in-wheel (service/foundation) D-Brakes have been designed and demonstrated.



D-BRAKE COOLING SYSTEM

A unique internal coolant flow configuration directs the coolant radially outward, allowing the coolant to flow at low temperatures and low pressures.

Figure 2-15 DeConti Brake Details

An example of the impact of the deConti system on braking performance can be seen in Figure 2-16, which is based on a series of braking tests performed on a dual-rear-wheel cube van at its maximum capacity of 11,200 lb. (5080 kg). Test procedures and conditions included dry pavement, ambient air temperature of 30°F (-1°C), and 40 lb. (18 kg) of brake pedal force to a full stop from 60 mph (96 km/h). Initial brake temperature was 200°F (93°C) immediately prior to the commencement of each series [18].

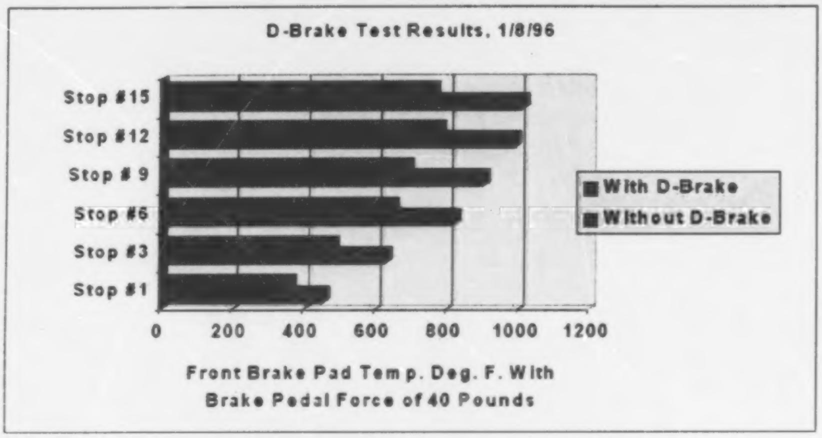


Figure 2-16 DeConti Brake Performance Comparisons

Another approach to improved cooling is the attachment of air movement enhancement devices. One such device has recently been developed by Stemco Inc. under the trade name of BrakeTrubine (as shown in Figure 2-17). The system is a retrofit attachment of a finned ring attached to the wheel end, which forces more air onto the drum to increase the cooling rate (and keep the drum free of debris).

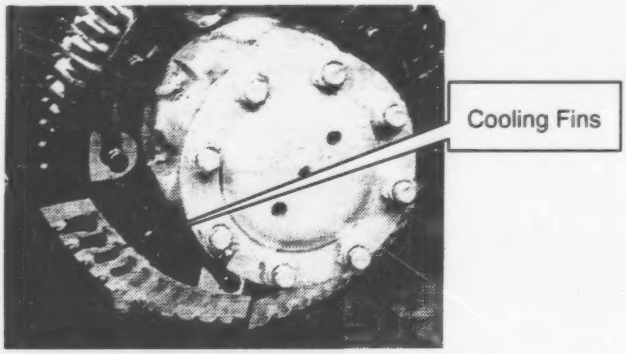


Figure 2-17 Brake Turbine

Full-Contact Brakes

Full-contact disc brakes are in general use on aircraft and industrial tensioning brake systems (Figure 2-18). While a caliper disc brake applies a pad clamping force over a limited area of the disc, the full-face-contact disc brake maximizes the potential contact area by using complete discs as both the static and rotating components. In aircraft, military vehicles and industrial applications, they are predominantly used in disc stacks, with several stators separated by rotors. During free rotation, the stator(s) and rotor(s) are separated. During application of the brake, a compressive load is applied across the disc stack and around the disc circumference. This load clamps the stator(s) and rotor(s) together, generating a frictional torque. This form of brake allows energy to be dissipated over a large friction area, which lowers the specific energy absorption rate, and thus heat and wear are significantly lower. This type of brake has been used on aircraft and large military vehicles from some years but the design has yet to be produced economically enough for the automotive industry.

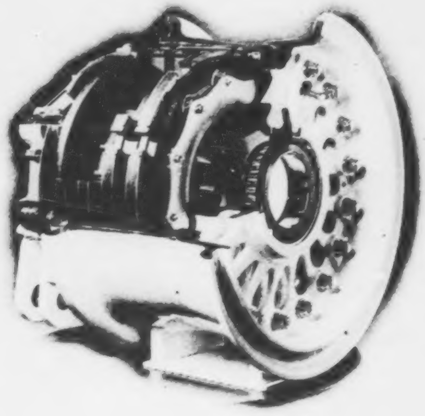


Figure 2-18 Example of a Full Contact Aircraft Brake System

NewTech Corporation, a Quebec-based company, has been developing an automotive version of the full-contact brake. This product has been under development for over five years, and was originally conceived as a truck product. However, the major development thrust for the past number of years has been on automotive applications – especially high-performance cars. The company, while having sold some product for OE evaluation, has yet to publicly disclose a supply contract with any automotive original equipment manufacturer (OEM). It has recently announced that it has re-established a truck brake development program with the involvement of Renault (Volvo Truck) [19]. A picture of NewTech's brake system is provided in Figure 2-19.

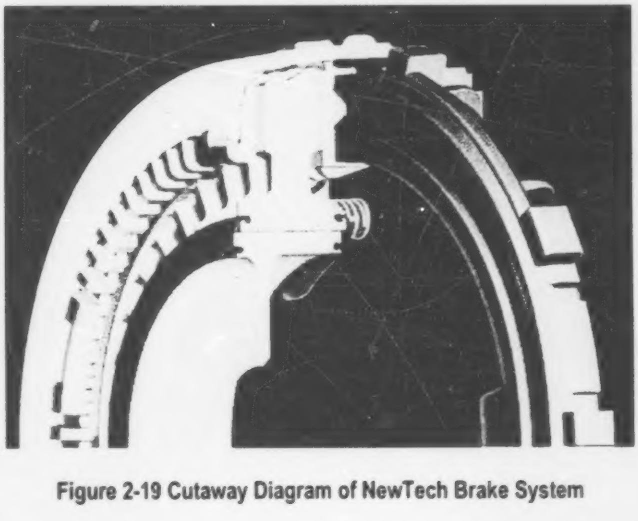


Figure 2-19 Cutaway Diagram of NewTech Brake System

Essentially, the NewTech brake consists of two annular pads: an outboard pad and an inboard pad. When the brakes are applied, these pads grip the entire surface of the rotor, distributing pressure over 360 degrees for maximum friction area. An aluminum driving hub connects the brake system to the wheel and provides easy access to the hub carrier.

Polymeric inserts located on the hub keep the rotor suspended while absorbing shocks. The rotor, made of a metallic composite, literally floats on the driving hub through the polymeric inserts. NewTech states that this special design not only eliminates all noise caused by resonating parts, but also helps ventilate the brake and dissipate heat through the cavities and fits on the rotor surface.

2.2.3 Other Wheel-End Components

The foundation brakes on a truck are part of the "wheel end", which includes the brake, wheel hub, wheel, and tire. This system must transfer all the vehicle weight to the road as well as transmit both acceleration and deceleration forces. Because the wheel end is "unsprung" – that is, on the "road" side of the vehicle's springs – any lightening of its mass will result in improving vehicle ride comfort and/or decreasing shock loading through to the frame or payload. It will also decrease the entire vehicle's mass so that payload can be increased and/or energy intensity decreased.

For these reasons, there have been a variety of alternate lightweight materials introduced to the wheel-end system over the last decade. It is instructive to review the developments in the other wheel-end components as they can shed light on the introduction of lightweight brake drums and rotors. All the new products have marketed themselves on multiple benefits, usually weight savings, increased durability or reliability, ease of maintenance, retrofit capability, etc. They all carry a cost premium.

Wheel Hubs

The principal suppliers of wheel hubs in North America are:

- ArvinMeritor
- Dana
- Hayes Lemmerz
- ConMet
- KIC/Reyco

In recent years, two new wheel hub products have been introduced to the market to compete against the cast iron product – the aluminum and the ductile iron hub.

ConMet is leading the aluminum option with its PreSet™ pre-assembled wheel ends. A key marketing element is the extended service period of the products. Thus, when installed as OEM equipment, PreSet™ Wheel Ends are designed to be maintenance-free for at least 350,000 on-highway miles (563,270 km). It also provides a "permanently greased PreSet", which offers an extended warranty of 5 yrs/500,000 miles (804,672 km).[6]

All PreSet™ wheel ends are assembled with a lightweight aluminum alloy hub with precision machining, bearing cups and cones with reduced width tolerances, a specially designed bearing spacer, an extended wheel life seal and an ABS tone ring (Figure 2-20). Components are easily field serviceable, compatible with all wheel mounting systems and braking applications, and are readily available and retrofittable as an assembly. They are available for all industry standard steer, drive and trailer axles, both tapered and "P" spindle, with a common, simple installation procedure for all positions.

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[6] Virtually all other manufacturers have also introduced pre-assembled wheel ends in recent years.

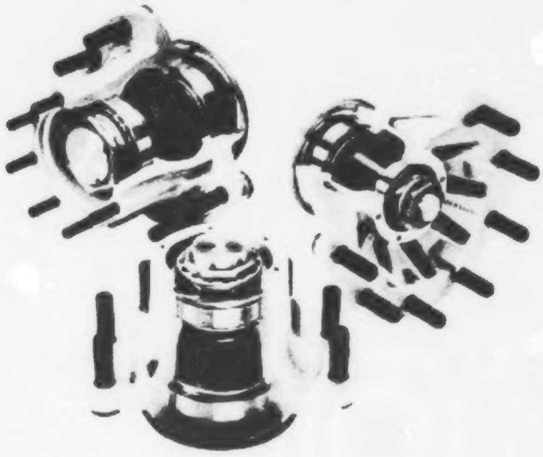


Figure 2-20 ConMet Aluminum Wheel Hubs

In competition to the introduction of aluminum hubs, ArvinMeritor and Webb Wheels offer an ADI hub that exceeds the tensile and yield strengths of aluminum (Table 2-3) and comes close to the weight savings of the aluminum product (Table 2-4).

Table 2-3 Material Performance Comparisons

Material	Tensile (PSI)	Yield (PSI)
Aluminum	41,000	30,000
Ductile Iron	65,000	45,000

Information from ArvinMeritor catalogue SP-9933

Table 2-4 Comparisons of Hub Assembly Weights

Manufacturer	Assembly (Cups and Studs) Weight (kg)
ArvinMeritor Standard	20.4
Competitor A	20.6
Competitor B	24.0
ArvinMeritor Ductile Iron	16.3
Aluminum	14.5

Information from ArvinMeritor catalogue SP-9933

Wheels

A similar materials battle has been waged in the wheel sector with standard steel fabricated wheels being challenged by both forged and cast aluminum products. The forged product is from Alcoa and has been on the market for over 10 years. The cast product is relatively new and is supplied by Hayes Lemmerz. Between them, the aluminum competitors currently have about 20 percent of the total North American and European wheel market.

As with hubs, marketing is targeted at weight reduction and durability. Table 2-5 provides information about the weight advantages of the cast aluminum product, which also has a higher heat transfer rate and thus stays cooler under repetitive braking situations. Data from Hayes Lemmerz (Speedline) [20] in Table 2-6 clearly show the thermal performance improvements with the cast aluminum wheel.

Table 2-5 Cast Aluminum Wheel Weight Saving Example

SEMITRAILER	WEIGHT SAVINGS (kg)
6 wheels + spare	- 133
12 wheels + spare	- 248
TRUCK	
6 wheels + spare	- 133
TRUCK+TRAILER	
12 wheels + spare	- 248
18 wheels + spare	- 363

Camozzi, 1999

Table 2-6 Thermal Performance Differences of Various Wheel Material Types

Number of Brake Tests	Cast Aluminum Wheel	Forged Aluminum Wheel	Steel Wheel	Cast Iron Wheel
	Maximum Temperature °C			
2 Brakings	287	290	285	285
% Change	\	+1	-1	-1
4 Brakings	453	486	440	456
% Change	\	+7	-3	+0.1
8 Brakings	627	690	667	650
% Change	\	+10	+6.5	+4
	Time in Seconds			
Time for brake disk to cool from 550°C to 200°C	1280	2013	1812	1575
% Change	\	+57	+42	+23

Camozzi, 1999

2.3 The Truck Brake Market

The market for brake drums/rotors (and all brake components) is divided into two broad areas:

1. The OE part market, where parts are specified, procured and installed by the OEM, who certifies the vehicle as meeting the new product standards of the country in which the vehicle is sold as well as meeting its internal quality and performance criteria;

2. The aftermarket, where the part is usually sold directly to the end user as a replacement or repair part and OE or new product standards do not apply.

The arrangement of the variety of suppliers, manufacturers and standards agencies in the marketplace is illustrated in Figure 2-21 [1].

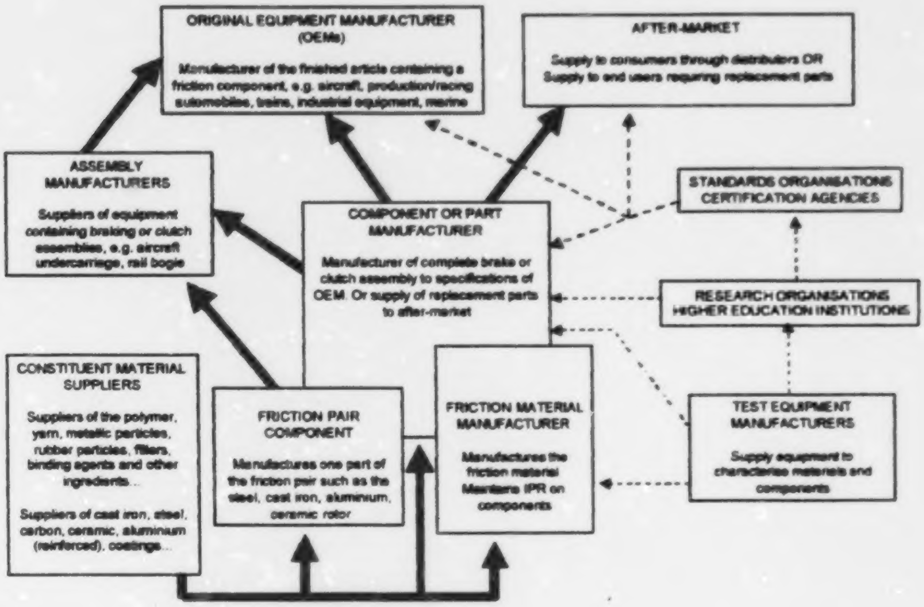

ORIGINAL EQUIPMENT MANUFACTURER (OEMs)
Manufacturer of the finished article containing a friction component, e.g. aircraft, production/racing automobiles, trains, industrial equipment, marine
AFTER-MARKET
Supply to consumers through distributors OR Supply to end users requiring replacement parts
ASSEMBLY MANUFACTURERS
Suppliers of equipment containing braking or clutch assemblies, e.g. aircraft undercarriage, rail bogie
STANDARDS ORGANISATIONS CERTIFICATION AGENCIES
COMPONENT OR PART MANUFACTURER
Manufacturer of complete brake or clutch assembly to specifications of OEM. Or supply of replacement parts to after-market
RESEARCH ORGANISATIONS HIGHER EDUCATION INSTITUTIONS
CONSTITUENT MATERIAL SUPPLIERS
Suppliers of the polymer, yarn, metallic particles, rubber particles, fillers, binding agents and other ingredients...
Suppliers of cast iron, steel, carbon, ceramic, aluminium (reinforced), coatings...
FRICTION PAIR COMPONENT
Manufactures one part of the friction pair such as the steel, cast iron, aluminium, ceramic rotor
FRICTION MATERIAL MANUFACTURER
Manufactures the friction material
Maintains IPR on components
TEST EQUIPMENT MANUFACTURERS
Supply equipment to characterise materials and components

Figure 2-21 Brake Market Players

Brake drums and rotors are wear parts that must be replaced periodically in the vehicle's lifetime; therefore, the market for these parts is large. The total number of vehicles, their annual use and the frequency of replacement per use define the total parts market.

The total numbers of heavy-duty trucks (Class 7 and 8) and trailers in Europe and North America are provided in Table 2-7. As can be seen in the statistics, the total vehicle population is much larger than the new vehicle sales, making the aftermarket the largest portion of the total market. In North America, it is estimated that approximately 9 million brake drum units are replaced each year. At the current price range of drums, this works out to an estimated market value of between US$700 and $900 million per year. The new vehicle market varies in direct proportion to the total truck population and is typically 6 to 8 percent of the aftermarket size, or US$40 to $70 million. The total market value is thus estimated at US$740 to $970 million per year.

Since the EU has nominally the same number of vehicles, it can be assumed that the number of replacement units per year will be similar to that of the North American market. However, the market value will be higher in Europe because of the higher average parts prices and the use of disc brake systems. It is thus estimated at US$1 to $1.2 billion per year.

Table 2-7 Truck and Trailer Populations

Vehicle Type (millions)	US/Canada		European Union	
	New Vehicle Sales	Total Registered	New Vehicle Sales	Total Registered
Truck/Tractor	0.33	3.05	0.30	~ 3
Trailers and Container Chassis	0.30	~ 3	~ 15	1.6

Data from OECD, US DOT, Transport Canada

While product acceptance and performance standards for new products are set by the OEMs and the governmentally-defined new vehicle performance standards, the aftermarket parts are less controlled. Also, while a large part of the market is supplied by OEM products, and thus have nominally the same performance as the new parts, the rest of the market is supplied by third parties that design a product to satisfy the marketplace. These products do not necessarily provide the same performance as the OEM products.

2.3.1 OE Brake Drum Suppliers

There has been significant corporate amalgamation in the automotive supply industry over the past decade and this has seen a significant reduction in the number of suppliers of OE brake parts market. In North America, the market for drum suppliers is split between the full brake system suppliers and those firms that only supply the drum as a part for integration by the OEM into the brake system. The number of full brake system suppliers has been reduced through corporate mergers to:

- ArvinMeritor (Lucas, Rockwell, Dayton-Walther)
- Honeywell (Bendix, Knorr-Bremse) - soon to be part of GE

There are more drum parts suppliers (some sell to the system suppliers as well), with the largest suppliers being:

- Hayes Lemmerz
- Gunite Corp.
- Webb Wheel

All of these suppliers offer full design, build, and support services to the vehicle assemblers and develop speciality applications in conjunction with the OEMs on a project basis. They are mostly large, publicly traded multinational corporations with substantial financial and engineering resources.

The following is a list of North American brake system suppliers, with contact information:

- **ArvinMeritor** 2135 West Maple Road, Troy, MI 48084 Phone: 248-435-1933, Fax: 248-435-9946, Web: http://www.meritorauto.com
- **Dana Spicer** PO Box 4097, Kalamazoo, MI 49003-4097 Phone: 616-553-1925, Web: http://www.dana.com
- **DeConti Industries** PO Box 2786, New Britain, CT 06051 Phone: 203-225-0720, Fax: 203-827-4475
- **Erie Wheels** 603 West 12th St., Erie, PA 16501 Phone: 814-452-6431, Fax: 814-456-0830
- **Euclid Industries** 6660 Beta Drive, Cleveland, OH 44143 Phone: 440-461-4300, Fax: 440-947-5998
- **Gunite Corp.** 302 Peoples Ave., Rockford, IL 61101 Phone: 815-964-3301, Fax: 815-965-9197, Web: http://www.gunite.com
- **Hayes Lemmerz** 15300 Centennial, Northville, MI 48167 Phone: 800-521-0515, Fax: 734-737-2071, Web: http://www.Hayes Lemmerz.com

- **International Marketing Services** 238 Hoover Blvd. #30, Holland, MI 49423
  Phone: 616-396-9802, Fax: 616-396-9804

- **KIC Holdings** 4109 Fruit Valley Road, Vancouver, WA 98660 Phone: 360-696-0561,
  Fax: 360-696-3132

- **Webb Wheel Products** 2310 Industrial Drive, S.W. Cullman, AL 35055
  Phone: 205-739-6660, Fax: 205-739-6246

In the EU, the market is significantly different, with more of the market supplied by directly designed and manufactured parts from the OEMs (Daimler-Chrysler, Volvo, Fiat, etc.). The OEM parts share is estimated at over 50 percent, but is shrinking as a result of cost pressures that are best managed through out-sourcing. The European Tier 1 parts suppliers (suppliers of a wholly assembled part or system) are also reducing in number through the same corporate conglomerations that have occurred in North America. However, there are a few more suppliers than in North America. The largest non-OE suppliers are:

- ArvinMeritor
- Hayes Lemmerz
- Brembo (an Italian brake designer)
- Knorr-Bremse (acts independently of Honeywell, with which it has a North American marketing agreement)
- BPW Berische (a German axle and brake manufacturer)

ArvinMeritor has the overwhelming lead in the North American market, with 80 percent of new vehicle brakes. This has been developed through its purchase of market share through corporate acquisition as well as its development of fully integrated axle-brake combinations. The European market is much less concentrated and ArvinMeritor is said to have only 10 to 15 percent of the new vehicle market. BPW and Knorr-Bremse together are estimated to control 25 to 30 percent of the new product market.

2.3.2 North American Aftermarket Industry

As the last two years of record-setting sales of new heavy-duty trucks now begin to cool down, the age distribution of trucks will now elongate slightly. This will increase the relative size of the aftermarket because of the longer service lives and because the new trucks that have been purchased in the past few years will start to need replacement components.

Traditionally, aftermarket components have been purchased through OE dealers and warehouse distributors (WDs). However, a third selling channel has recently entered the picture - buying groups.

WDs: A WD is a company that buys components from manufacturers at reduced prices, then distributes the products to independent repair facilities. There has been a trend for large WDs to acquire smaller ones and grow even larger. The resulting larger distribution network provides larger WDs with more access to market, and a correspondingly larger market share.

OE Dealers: While the primary focus for dealers remains new truck sales, they have also recently become more involved in the replacement parts business.

Buying Groups: A buying group is a conglomerate of WDs that buys components from manufacturers at discounted prices, then distribute the product through the group's members. Buying groups are expected to become increasingly influential in how fleets and operators purchase their products.

Although the Canadian aftermarket is similar to that in the US, there still remain significant differences:

- Canada's vast geographic area and widely dispersed population centres often force fleets and drivers to scramble for parts while on the road. This means that suppliers who want to continue growing in Canada must open distribution centres throughout the country with low sales density networks.
- In general, there is a relatively larger market for heavier duty components in Canada because of the wide diversity of applications. Over the past decade, there has been a rapid increase in the installed horsepower rating of truck engines (this analysis is based on large trucks). Data on Cummins engines [21] in Figure 2-22 show this increase, and also illustrate the fact that Canadian trucks have larger engines than US trucks. This higher rating is a result of Canada's higher size and weight limits and, to a lesser degree, its more severe winter temperatures. The increase in horsepower will tend to increase fuel consumption rates – the amount of the increase is highly variable depending on whether the extra horsepower is used to haul more load, drive at higher speeds or climb hills faster.
- Canadian economics and tax laws have resulted in an older population of vehicles on Canadian roads, as can be see in Table 2-8 [22]. This means that there is a proportionally larger aftermarket than in the US.

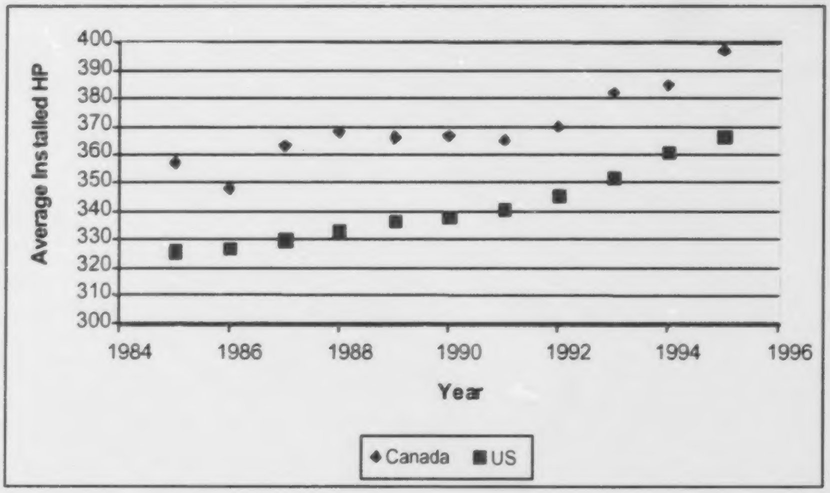


Figure 2-22 Installed Horsepower Trends

Table 2-8 Comparison of Equipment Ownership Life

Country	Mean Age of Tractors (years)		Mean Age of Trailers (years)	
	Truckload	Less Than Truckload	Truckload	Less Than Truckload
Canada	6.8	5.4	9.0	8.5
US	3.0	7.0	5.7	12.5

For-hire operators only

2.3.3 Aftermarket Brake Drum Suppliers

All the OE brake drum suppliers also participate in the aftermarket. They do this either directly or through the vehicle manufacturers' parts supply channels. Their parts tend to be at the high end of the price distribution, perhaps because of their higher engineering overheads, quality assurance and warranty costs. Because the aftermarket does not have the OE quality "filter" for parts quality and performance, it is also supplied by firms that do not sell to the OEMs and thus the design standards may not be as high as for new vehicle parts.

Drum suppliers do not generally cast a large portion of their product, but source raw castings from casters throughout the world. The parts "manufacturers" then provide quality assurance, final finishing, product distribution, technical support and warrantees for the cast product. This industry structure has developed as a result of the availability of low-cost aftermarket castings from third world countries. These products have dropped the price bottom out of the market and created severe price competition in the marketplace. Current "no name" cast iron replacements can be purchased for as little as C$70, each with "OE quality" products up to twice this price.

2.3.4 Significant Market Differences

The North American and European marketplaces are significantly different in a number of ways:-

- The buyers specify the brake system components (or supplier) in North America when ordering a truck or trailer, whereas in the EU the OEM has total control of the brake system design.
- EU brake standards are different, having essentially forced a shift to disc brakes because of higher energy loading demands, faster actuation times, etc.
- There are more small fleets in North America compared to the EU, and thus less central fleet buying. There are also fewer engineering resources for product specification or testing.

2.4 Market Value for Weight Reduction

The goods movement market is capable of getting value out of vehicle tare (empty) weight reduction in two ways:

1. through decreased total vehicle mass, which decreases fuel consumption; and
2. through increased payload, which increases revenue or decreases costs.

This section takes a look at what the North American market is currently paying for weight reduction as a basis for assessing what it would pay for lighter weight brake systems.

Vehicle mass is a critical variable in determining fuel consumption. While the exact relationship varies from one situation to another (truck configuration, relevant weight range, road surface/condition, tires, wind, speed, road vertical geometry, etc.), the relationship is probably within the range of 0.02 to 0.06 km/L for every tonne of weight saved or reduced. For example, a typical heavy diesel truck decreases fuel consumption in a range of 1 to 3 percent for every decrease of one tonne.

From the series of interviews undertaken as part of this study, a segmented market model seemed to emerge from the consensus opinions of the engineers, marketing managers and operators. This model is portrayed graphically in Figure 2-23 and shows that there is a high adoption (more than 90 percent) of weight saving options when the cost of these options is less than C$3/kg. In the conventional trucking market the economics are strictly a function of the duty cycle – therefore fuel saving potential – and vehicle usage that combine to provide the annual cost savings. These savings are then compared with the capital costs, usually by simple payback time or net present value analysis, to complete the buy decision. As the truck usage increases (a smaller percentage of the total market) larger cost-per-kilogram reductions can be justified. Therefore, as indicated in Figure 2-23, the potential market is progressively smaller.

There is, however, a second market: that of the carriers of high-value product or product that can be incrementally increased, and the vehicles that are "grossing" out at present. This segment of the market includes liquid haulers (especially high value liquids like gasoline), concrete haulers, and resource haulers (wood chip, gravel, grain, etc.). Because this market makes up less than 20 percent of the total vehicle market and can extract a secondary cost saving from tare weight reduction, it can justify a higher cost for weight reduction. These are niche applications that can support the first introduction of new weight reduction products.

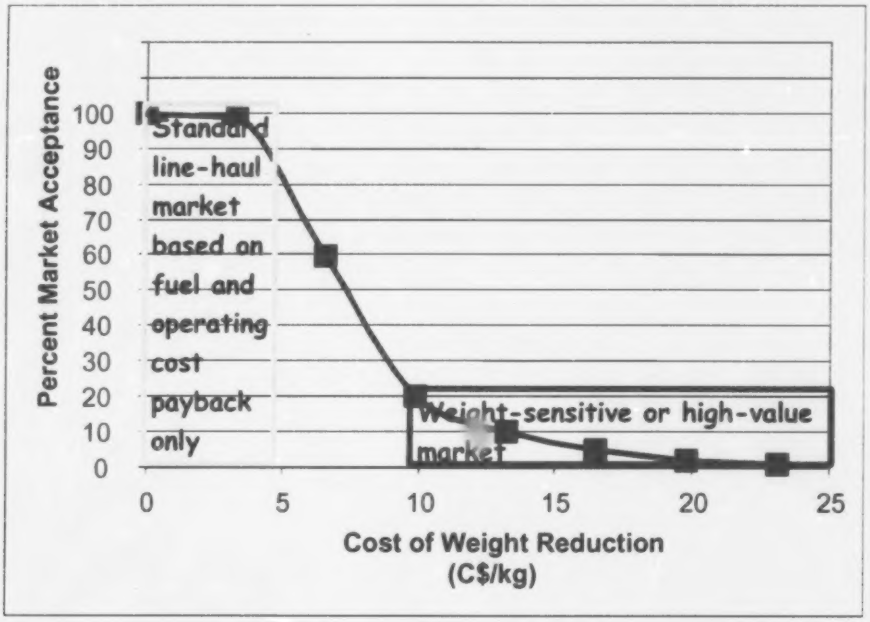


Figure 2-23 Market Penetration versus Weight Reduction Cost

2.5 Tare Weight Reduction Options

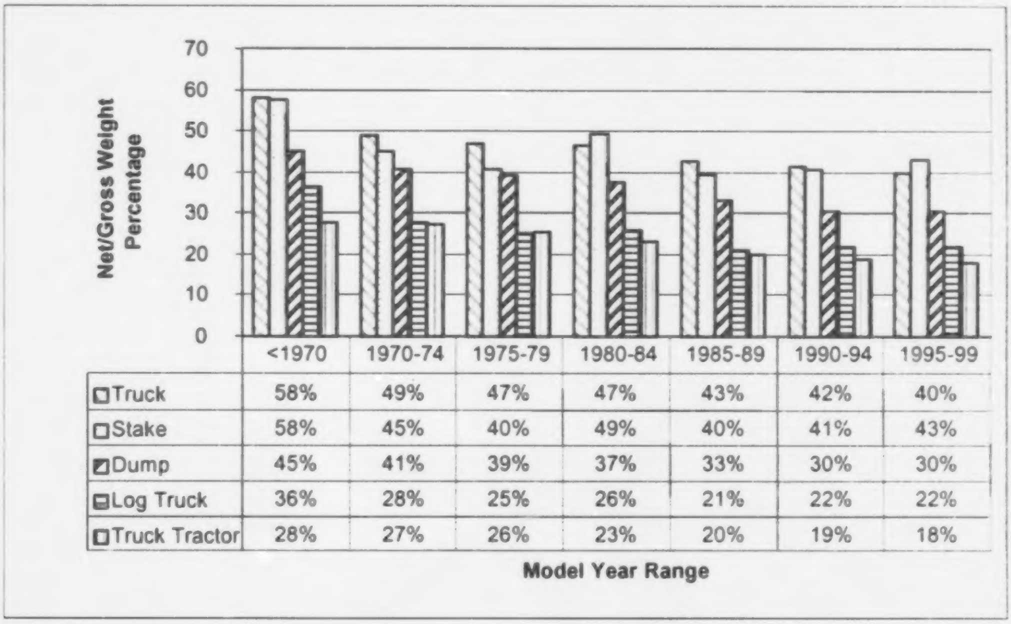
There are two key issues associated with the tare weight of a truck:

1. If tare weights can be reduced for trucks in general, fuel requirements will decline simply because the amount of energy required to overcome inertial forces and rolling resistance will decrease.

2. For the approximately one-in-five truck trips that gross out[7] – that is, the payload fills the weight capacity of the truck – a reduction in truck tare weight makes heavier payloads possible. In this situation, the same volume (tonnes) of freight can be hauled with fewer truck trips and, hence, less energy.

There is a great deal of variability in truck tare weight, but where a carrier needs light equipment and can operate in a suitable environment, trailer manufacturers can generally build light trailers and truck/tractors can be found that are 1000 kg or so lighter than many on the road. There has been a general trend to high structural efficiency (tare weight/registered GVW), as can be seen in Figure 2-24, which was created using Insurance Corporation of British Columbia data files.

The weight saving has been achieved through design optimization to reduce mass or through material substitution. In some payload-sensitive applications, such as petroleum haulers and log trailers, and in some specific markets, such as California, there are now trailers that are almost totally aluminum. The exceptions are limited to the axles, brakes and a few other parts. But this high weight sensitivity is restricted to a small portion of the total truck market – estimated at 10 to 15 percent.



	<1970	1970-74	1975-79	1980-84	1985-89	1990-94	1995-99
▨ Truck	58%	49%	47%	47%	43%	42%	40%
☐ Stake	58%	45%	40%	49%	40%	41%	43%
▨ Dump	45%	41%	39%	37%	33%	30%	30%
▤ Log Truck	36%	28%	25%	26%	21%	22%	22%
☐ Truck Tractor	28%	27%	26%	23%	20%	19%	18%

Figure 2-24 Historical Changes in Structural Weight Efficiency

To give an idea as to how far this push for greater structural efficiency can go, a bulk hauler in the US (80,000 lb. or 36,287 kg regime) is now able to haul payloads of 56,000 lb. or 25,401 kg. A decade ago, the conventional rule of thumb was that 45,000 lb. (20,142 kg) payloads were the norm under the US federal 80,000 lb. cap.[8]

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[7] This statement is based on the 1995 CCMTA (Canadian Council of Motor Transport Administrators) survey, in which 19.4 percent of the truck trips on main highways across Canada in a one-week period were loaded to the weigh-out (or gross-out) point. This calculation excludes trucks with unknown load status.

[8] The material in this section was developed from a wide range of background information and has benefited from discussions with a number of manufacturers or operators, including Trimac Transportation System, Innovative Trailers (TX), BWS Manufacturing (NB), and Ty-crop Manufacturing (ON).

Just because it is possible to operate light equipment does not mean many trucks are actually in this category. There are many reasons, including:

- Trucks may be used in rough operating conditions (construction, logging) where the operator chooses heavier, sturdier equipment for handling and maintenance reasons;
- Heavier tractors than necessary are used because of unrelated choices made by the operator.

Regarding the latter point, one Canadian carrier suggests that when preparing a bid for major hauls, equipment should be carefully specified to maximize payloads. In doing this, they purchase trailers as light as possible and design their business economics around a tractor weighing 8172 kg (18,000 lb.). The difficulty is that they cannot find owner-operators with tractors under 9080 kg (20,000 lb.) The dilemma, then, is that the extra 908 kg (2000 lb.) on the tractor reduces payloads by an equivalent amount. Given that owner-operators live in these tractors for long periods, however, the company has no choice but to use heavy tractors.

The amount of variability in tare survey weights in vehicles on Canada's roads can be illustrated by data from the 1995 CCMTA Roadside Survey (Table 2-9). The distributions shown are based on an estimate of truck trips where cargo weight is zero. Truck weight is only given in ranges (i.e., not a continuous variable). Some of the "coefficients of variation" in these estimates are high, but this is not important here because the interest is not in the absolute value of the estimates. Some of the extremes in the values shown are accounted for by such things as specialized equipment or trucks with certain equipment (e.g., five-tonne loading cranes) attached.

Table 2-9 In-Service Tare Weights Distributions

Weight Range (tare) tonnes	5-axle tractor-semitrailer	6-axle tractor-semitrailer	7-axle double	8-axle double	3-axle tractor
3-5					0.6%
5-10	0.04%				92.5%
10-15	33.6%	3.6%			5.7%
15-20	61.9%	81.9%	61.2%	28.7%	1.2%
20-25	4.3%	12.4%	34.6%	66.5%	
25-30	0.1%	2.1%	4.2%	3.2%	
30-35				1.0%	
35-40					
40-45					
45-50					
50-55				0.5%	

So, for example, while most five-axle tractor-semitrailers (tanks, flats, vans, etc.) have a tare weight in the range of 15 to 20 tonnes, it is possible to operate this equipment in the 10 to 15 tonne range and there even are a few (0.04 percent) that weigh less than 10 tonnes.

Unfortunately, because of the wide variety of truck types captured in these figures (a container chassis hauled in a five-axle tractor-semitrailer configuration is considerably lighter than a double-walled, stainless steel milk tank) and because of the use of weight ranges in the CCMTA survey, it is not possible to use these data to make meaningful statements about the current market share for lightweight equipment. However, given that about one in five truck trips (including the empty ones) grosses out (so tare weight is important for these carriers), it may be fair to assume that a large proportion (say one third to one half) of the trucks are purchased for hauling situations where tare weight is an important issue in the sense that it affects payload capacity.

Factors affecting the differences between a "heavy" and a "light" truck - aside from the obvious things such as the differences in the number or length of trailers and the number of axles - are many. A partial list, with a rough indication of the weight that can be saved by choosing various features, follows in Table 2-10 (the rough indication of weight uses as its base a five-axle tractor-semitrailer specified for US operations).

Table 2-10 Typical Weights of Truck Optional Components

Components	Typical Weight Differences
air dryer	12 kg
aluminum air reservoirs	8 kg
aluminum radiator	12 kg
choice of wheels	up to 200 kg (or more on multiple axle configurations)
choice of tires	up to 300 kg (or more on multiple axle configurations)
rear axle suspension/brake/etc.	up to 275 kg
front axle suspension	up to 54 kg
trailer suspensions	up to 275 kg (or more on multiple axle configurations)
composite springs	up to 363 kg (or more on multiple axle configurations)
fixed fifth wheel	45 kg
manual slide fifth wheel	up to 20 kg
aluminum fuel tanks	up to 72 kg
driver's seat	20 kg
tilt steering wheel	9 kg

2.6 Cost of Weight Reduction

Some reductions in tare weight reduce the cost of equipment - for example, specifying a truck with a fixed fifth wheel, smaller sleepers, a shorter wheelbase, standard driver's seats, or an engine that is not overpowered. Others increase costs - for example, choosing more aluminum in the trailers. The range of material substitution costs can be seen by looking at some typical option or alternative costs as provided in Table 2-11. From this data, the margin price premium per kilogram of weight saved can be derived. The least expensive substitution of these examples is changing the sidewalls of a van trailer from steel to aluminum, which has a marginal cost of C$6.60/kg of weight saved (a typical unit is shown in Figure 2-25). The next least costly is the combination of aluminum and steel in a flat deck trailer, which costs C$10.80/kg of weight saved. Moving to an aluminum-intensive flat deck or chip van has a marginal cost of C$12 to $15/kg change, while the most expensive substitution is for aluminum wheel at C$23.35/kg of weight saved.

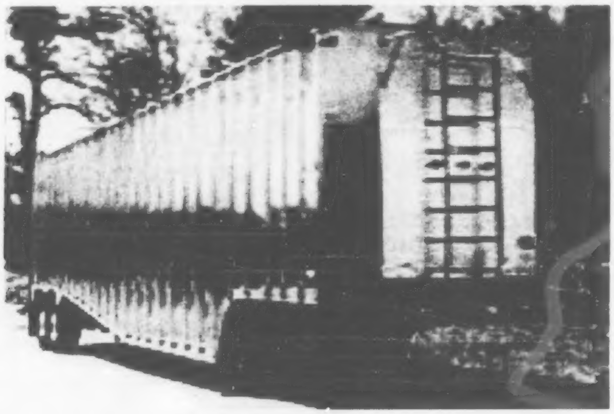

Figure 2-25 Typical Wood Chip Van Trailer

Of these comparisons, the wheel example seems to be out of line. This may be explained by the fact that aluminum wheels are usually sold with a high gloss polish and thus marketing of this product is driven by the "look good" value – which the market will pay a premium for – as much as the weight and/or energy savings.

Table 2-11 Comparisons of Current Cost Differentials

Supplier	Product	Base	Alternative	Difference
A	*Flat Deck*	*Steel*	*Combo (St/Al)*	
weight (kg)		5,227	4,727	500
cost (C$)		21,300	26,700	5,400
C$/kg		4.07	5.65	10.80
A	*Flat Deck*	*Steel*	*Aluminum*	
weight (kg)		5,227	3,864	1,364
cost (C$)		21,300	38,000	16,700
C$/kg		4.07	9.84	12.25
A	*Van*	*Steel Sidewall*	*Al Sidewall*	
weight (kg)				136
cost (C$)				900
C$/kg				6.60
B	*Chip Van*	*Conventional*	*Aluminum*	
weight (kg)				3,000
cost (C$)				45,000
C$/kg				15.00
C	*Wheels*	*Steel*	*Aluminum*	
weight (kg)				96
cost (C$)				2,250
C$/kg				23.35

2.7 Weight Impact on Fuel Use

There are a number of sources of information on the relationship between truck weight and fuel consumption. For example, from data in the 1998 edition of the American Trucking Association's (ATA) *Fleet Manager's Guide to Fuel Economy* [23], a fuel consumption-gross weight curve was developed as shown in Figure 2-26. It indicates a low variation in the sensitivity over the top 25 percent of the maximum weight of between 0.7 and 0.9 percent reduction in fuel consumption for every 1,000 kg reduction. This is for long-haul operations typical of ATA's members.

Several Canadian empirical studies have been done and the results of this research are provided in Table 2-12. It can be seen that the fuel consumption has been steadily decreasing over time. The sensitivity of fuel consumption – the change in L/100 km with a change in mass – has a relatively small variation. Over the entire period the sensitivity averages 1 percent.

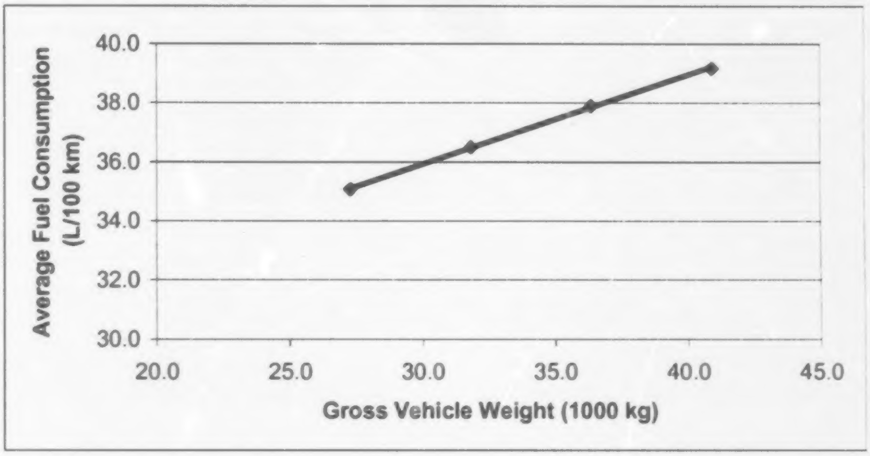

Figure 2-26 Fuel Consumption versus GVW at 120 km/h

Table 2-12 Truck Fuel Consumption Study Results

Data Reference	Average L/100 km	% Change /Tonne
Wilson, 1975/76 data on Day & Ross tractors [24]	45.0	1.1
Clayton, 1981-84 data on 13 carriers [25]	37.7	0.8
Nix, 1989 Ontario "DriveSave/TruckSave" [26]	28.8	0.6
Nix, 1991 Manitoba "Pro-Trucker Challenge" [26]	23.5	1.0
Nix, 1993 Manitoba "Pro-Trucker Challenge" [27]	23.9	1.0
Nix, 1994 Manitoba "Pro-Trucker Challenge" [27]	23.9	1.5
Nix, 1995 Manitoba "Pro-Trucker Challenge" [28]	20.1	1.1

Another way to estimate the fuel impact is to look at the sensitivity of fuel use to weight in vehicle fuel consumption models used by the major engine or vehicle manufacturers. Cummins Engine Company's Vehicle Evaluation Vehicle Mission Simulation model is a good example and was used to model the effects of both gross weight and duty cycle on fuel use. The vehicle was defined as a 22,000 kg GVW dump truck (6X4) with a 370 hp engine installed. This "vehicle" is also similar to the specification for an urban bus (without the extra axle, the elimination of which provides more passenger capacity or payload), which has the same range of GVW (curb weight about 10,000 kg) and has between 270 and 300 hp installed. The output data is presented in Figure 2-27, which compares estimated changes in fuel consumption with a 1000 kg decrease in GVW for four driving cycles simulating different mission types:

- Urban bus cycle
- Urban goods delivery cycle

- Rural goods delivery cycle
- Long-distance interstate (flat to rolling terrain) cycle.

The largest change in energy use occurs with the urban bus cycle as it has a high frequency of stops and starts in which the effect of the gain and loss of the vehicle's inertia is the greatest. Operating the vehicle on this cycle results in a 3.4 percent change in fuel use with every 1000 kg change. The urban delivery cycle is the next most sensitive with a 2.4 percent change per 1000 kg and the rural delivery is similar at 2.1 percent. The interstate route shows the least effect of GVW change at 1.2 percent change in the fuel consumption rate (L/100 km) for every 1000 kg change in the vehicle's GVW. This final value is very close to the empirical data presented in Figure 2-26 and Table 2-12.

The economic value of these changes depends on the price of fuel and the relative amount of driving that a fleet does per year in the various mission types. As an indicator of the value of the fuel consumption changes, the fuel was valued at C$0.70/L. The possible financial savings are thus C$1.33, C$1.06, C$0.74, and C$0.30 per 100 km for every 1000 kg reduction when operated on the urban bus, urban delivery, rural delivery, and interstate routes respectively.

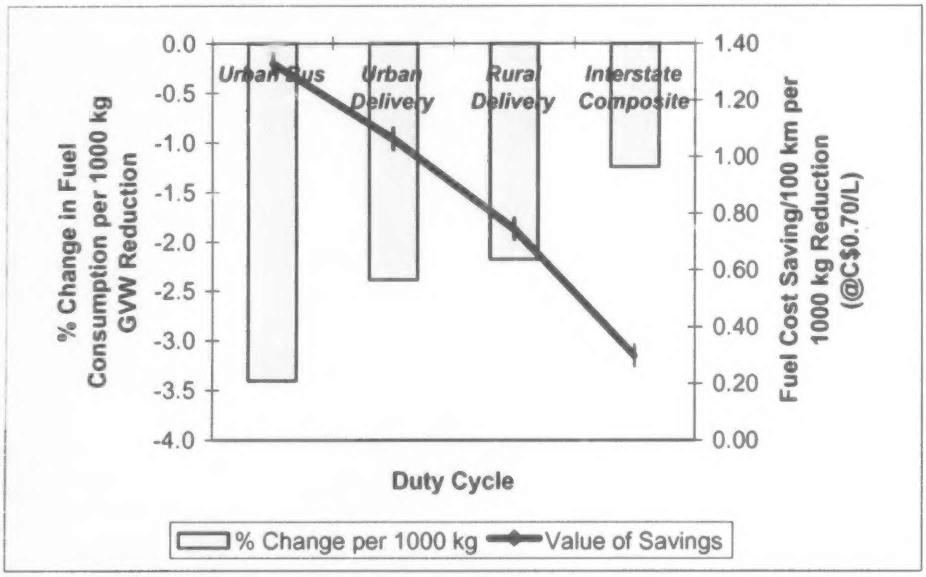

Figure 2-27 Influences of Weight and Duty Cycle on Fuel Consumption

2.8 Value of Brake Wear Reduction

One of the secondary characteristics of both MMC and C/SiC materials is the possibility of greatly reduced component wear rates. The importance of this factor lies in two effects:

1. A decrease in the frequency of the need to replace worn parts reduces operating costs through lower labour and parts costs, and through reduced vehicle downtime so that productivity is improved.

2. A decrease in drum (or rotor) wear rate directly affects the frequency of brake adjustment which, if reduced, increases the probability that brakes are within adjustment tolerances and can thus operate more effectively.

2.8.1 Maintenance Cost Value

The cost of brake maintenance is one of the highest operating costs for a truck or bus. Various sources have estimated that brake maintenance costs account for the largest percentage of maintenance costs and that brakes have the shortest time between repairs or adjustments. The rate of brake repair is of course highly dependent on the total amount of a vehicle's use as well as the characteristics of its use (e.g., stop and go, mountainous, light goods, heavy goods, etc.). There is a great range of applications for trucks and thus large differences in these factors. Three main usage groups have been typically defined by the industry as:

1. long haul with over 150,000 km/y,
2. mixed haulage at 100,000 km/y, and
3. short or local haulage at 50,000 km/y.

The most onerous duty cycle for brake wear is probably the urban transit bus, while the long-distance dry goods truck has the most benign mission for brake wear. Data from a European disc brake manufacturer [29] illustrates the link between usage pattern and annual distance travelled, and service life expectancy of pads and discs.[9] Figure 2-26 shows, for example, that 80% of the long-haul fleet (i.e., the fleet travelling more than 150,000 km per year) experiences a brake pad service life of more than 500,000 km, and that 20% of this same fleet experiences a service life of more than 1,200,000 km.

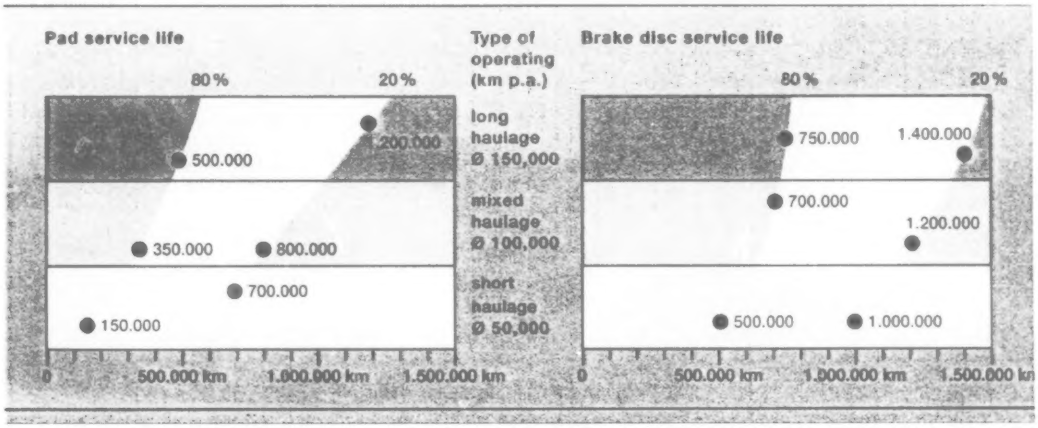

Figure 2-28 BPW Disc and Pad Service Life Curves

As brakes are used, the components within them wear. The highest wear rate is for the friction pad material, followed by the drum/rotor. The current market is largely a "use and replace" market and should remain that way for the foreseeable future. Typically, a drum is replaced at every second pad replacement. However, in some operations the drum may be machined or "turned" to bring it back into round and used for a third pad cycle. This practice of drum turning has been decreasing over the last decade as a result of a variety of market pressures, including labour cost increases, labour supply shrinkage, drum price decreases, etc.

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[9] From conversation with truck operators in the UK, the absolute values in this chart are probably too high but the relative performance trends are correct.

The cost of brake drum overhauls was estimated in 1994 through fleet surveys [30] as C$600 to $700/year per unit (3-axle truck or trailer, or 2-axle trailer) or an average of C$125 per wheel end per year. This cost does *not* include the periodic costs of brake inspection, adjustment and cleaning, which can easily double this cost.

In transit applications [31], these costs are higher, as it is typical for a bus to require two brake overhauls per year, resulting in brake overhaul costs alone of between C$3500 and $4000 per bus per year.[10] The drums on transit buses are normally machined and reused at least once. Thus, the replacement brake drum costs are approximately one tenth of the total annual costs. With the MMC material, machining of the drums is not likely; however, with wear rates potentially less than one third that of conventional drums, the frequency of relining should decrease in proportion, providing maintenance cost savings throughout the bus's life.

2.8.2 Brake Adjustment Reduction

Changes in the wear rates of brakes directly affect the frequency at which brakes must be adjusted. As brakes are used and as pad and drum material is lost, the distance between the two materials increases, which causes increased delay in the actuation of brakes and decreased maximum total braking force. The "slack" that is developed is either adjusted out by mechanical adjustment of the pushrod (see Figure 2-29) or through the use of an automatic slack adjuster (ASA). In the case of the manual system, adjustments must be made daily when vehicles are in severe duty usage. The ASA greatly reduces the manual adjustment requirement but even it can have reliability problems, which lead to brakes being "long" and thus beyond the legal limits for on-road safety inspections. As brake stoke adjustment is the most common reason for which trucks are put out of service, any technology that can reduce the probability of this out-of-adjustment problem would be well received by both industry, which would have less downtime and fines, and governments, because trucks would be closer to original specifications more often and thus safer.

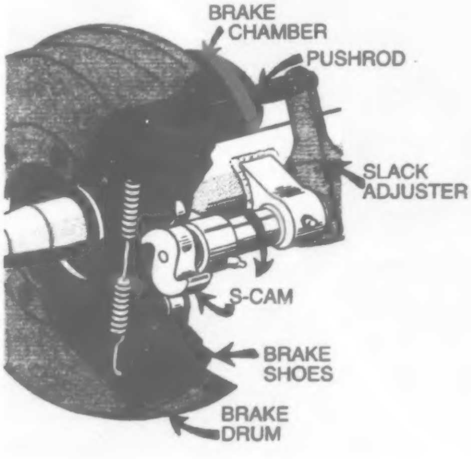

Figure 2-29 S-Cam Brake Components

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[10] Information on brake maintenance costs was obtained from municipal transit authorities in Toronto, Ottawa, and New York City.

The fact that MMC and CCC materials have the potential to greatly reduce the wear rates is thus a very important physical attribute of the materials. The magnitude of the impact of changing the wear rate has been assessed in a prior study [32]. It was estimated that for a single or dual drive axle truck (or bus), reducing wear rate by 25 or 50 percent (increasing time to replacement by 50 to 100 percent) could reduce the likelihood of brakes being out of adjustment by 80 to 83 percent (see Table 2-13). In comparison, the ASA (which is now mandated as standard equipment) reduces the out-of-adjustment probability by 75 percent.

This change in wear rate is well within the potential effect of the material change and provides strong evidence that brake performance, on average, could be greatly improved by the use of low wear-rate materials.

Table 2-13 Effect of Brake Wear Rate Change on Brake Out-of-Adjustment Probability

Technology Change	Percent Change in Out of Adjustment Probability from Baseline (no ASA) System
ASA	-75
50% Wear Time Increase	-80
100% Wear Time Increase	-83

- based on a single or dual drive axle Class 8 straight truck or bus

2.9 Other Market Issues and Drivers

2.9.1 OEM Warranty Demands

OEMs are requiring extended warranty periods with a shift to offering leasing agreements to customers for the complete vehicle package. This is resulting in cost of ownership being important not only to the end user but also to the vehicle manufacturer. Minimum servicing is required with a typical duration of four years for this type of lease. The emphasis is therefore on robust drums/rotors that do not crack, and on friction material that, while meeting performance requirements, gives good life and comfort. This has led suppliers to the evaluation of new friction couples such as CCC, aluminum MMC and ceramics for truck applications, although to date grey cast iron remains the current production material for rotors. Initial cost and durability over a wide range of operating conditions are also important factors in the product material decision.

2.9.2 Product Cost Reduction

Comments from some of the OEM representatives interviewed indicate that one of the market "drivers" of the move to discs in Europe was the ability for the OEM to outsource the new (disc) brake system components. However, they could not have done so with conventional drum brakes because of labour retention laws, which are very strong in Europe. In the European market, a much larger percentage of the brake components are supplied by OE internal sources. These parts are generally more expensive than the outsourced parts because of production inefficiencies and higher labour costs. Therefore, while disc brakes were introduced in North America in the late 1980s, they had a significant price premium. The current generation of designs being offered in North America still carries a price premium. In Europe, the same cost structure results in a slight but significant cost saving (less than 10 percent) to the OEM as the base price of the drum product is relatively higher.

Thus, if the new material designs can play into this outsourcing market in Europe, then they will have an economic advantage over internal manufactured parts. In North America this is not the case, as virtually all brake system components are sourced from external suppliers now.

2.9.3 Vehicle Performance Regulations

The design of brake systems is primarily driven by government-imposed performance regulations that set the minimum sizing, function and performance. The market appears to have little interest in enhanced performance, although there is a market for enhanced durability and thus extended-wear brake systems and low-maintenance systems are coming on the market, such as the Camlaster system from ArvinMeritor [33].

The difference in regulated braking requirements between US (and Canadian) and EU new product regulations has played a significant role in shifting the EU market to discs and electronic braking systems. EU performance regulations emphasize high-energy braking, braking stability and fade minimization more than the US regulations and these demands are better met by disc brakes, which have inherently less fade as well as fast application and release times. This pressure has caused the EU new truck brake market to shift within the past five years to approximately 80 percent discs on tractors and straight trucks and 50 percent on trailers. In North America, discs account for less than 1 percent of the market.

There is clear demand by the general public, regulators and the industry for trucks that are safer and this is driving the regulators to move the braking performance requirements of trucks closer to those of cars. This will require faster actuation, which will move the systems to electronic actuation instead of the current air actuated systems. The systems will also need larger braking capacity. Both these trends will increase the likelihood that disc systems will be the brake system of choice (or need) in North America as well as the EU.

2.9.4 Product Distribution and Support

One key factor in a product's market success is the availability of both product and technical support after the initial sale. While this statement may seem entirely obvious, lack of replacement product or good technical support played a significant role in the lack of success of wedge brakes and disc brakes in the 1980s. The industry does not tolerate downtime due to parts unavailability and some owners have stated that they have often switched back to a less desirable part - for example, from composite drum to cast - because of the lack of availability in all areas of their operating territory. Any introduction of a replacement technology will have to either have a well supplied supply chain in the aftermarket system or be able to initially restrict the range of application of the drums/rotors to those areas that have acceptable product support.

2.10 Concluding Remarks

The brake systems of trucks are distinctly different in their requirements than those of passenger cars in that they are required to absorb and dissipate much higher rates of energy. This means that the maximum temperatures for truck brakes can be significantly higher than those of cars and this presents physical boundaries that may be difficult to meet for any new materials. While most of the vehicles on the road operate with drum brake systems, this technology has rapidly been replaced in the European market by disc brake designs, which have even higher specific energy loadings and operating temperatures. It is expected that disc brakes will attain a market "breakthrough" in North America only if regulations are changed to require their superior performance capabilities.

The market will accept the application of lightweight brake technology, once its technical capability is proven, on its direct economic value to the operator. The benefits lie primarily in the value of weight reduction. Current evidence from the marketplace suggests that about 20 percent of the market is very weight sensitive and will pay up to C$10/kg weight saved for weight reduction. Some smaller niches will pay even more. If the differential costs can be reduced to C$3/kg, then significant penetration of the mass truck market could be expected.

F2

The impact of weight saving on fuel consumption is a function of the duty cycle on which the vehicle is used and its configuration. Empirical and engineering model evidence suggests that line-haul operations would have the smallest impact at 1.2 percent change in consumption per 1000 kg reduction. Straight trucks in urban delivery operations and transit buses are expected to achieve 2.4 and 3.4 percent consumption reductions per 1000 kg weight reduction, respectively.

3. PHYSICAL ANALYSIS

This chapter provides an analysis of the physical capabilities of the various candidate substitute materials. The information in the chapter has been extracted from a separate MTL report on composite brake rotors [34].

3.1 Review of Patents

Since the early 1990s, numerous patents on the making of metal matrix composites have been filed. The patented processes of making MMCs using an inexpensive molten metal process can be classified into three categories:

1. Stir casting, whereby the reinforcement is fed into a container filled with molten metal under vacuum and a specially designed stirrer is used to mix the reinforcement homogeneously in the melt. The mixed melt is then cast to form composite ingots.
2. Using an infiltration enhancer precursor or reactive atmosphere to enable molten metal to spontaneously infiltrate a permeable mass of reinforcement or preform to form a composite. The reactive atmosphere systems that exhibit spontaneous infiltration are:
    - aluminum/magnesium/nitrogen
    - aluminum/strontium/nitrogen
    - aluminum/zinc/oxygen
    - aluminum/calcium/nitrogen [35]
3. Squeeze casting, which is very similar to the process used in the second category. Molten metal is infiltrated into a preform, with an application of mechanical pressure rather than a reactive atmosphere. Very often, a pressure of up to 100 MPa is used in the squeeze casting process.

Over the past ten years, many patents on improving various performance aspects of brake drums and rotors[11] have been filed and are summarized in Table 3-1. In addressing the high temperature service environment, the selected rotor materials need to have a good creep resistant property. Therefore, rotors made with stainless steels, titanium alloys, nickel-based alloys, iron-based alloys and copper-based alloys have been proposed. However, although the employment of these alloys for brake drum or rotor applications might satisfy high temperature requirements, necessary cost advantages and weight savings are not realized.

An alternative means of ensuring that rotors can endure the high temperature service environment is to apply coatings of iron-based alloys, alumina, intermetallic, or zirconate on the cast iron rotors. Again, despite the achievement of high temperature properties, there is no benefit in either weight or cost savings. Further, decohesion of coatings from the rotor surface is a prominent problem for some coatings.

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[11] The term rotor is used to mean the rotating mass of both disc and drum brake designs.

Another suggestion involves the incorporation of ceramic reinforcements into iron and copper-based alloys to form composites. Apparently, the combination of ceramic reinforcements and high melting temperature matrix alloys offers benefits in wear resistance and improved high temperature properties. Nonetheless, neither the weight savings aspect nor the cost is addressed.

Table 3-1 Selected List of Brake Rotor Patents

U.S. Patent Number	Author	Title
5325941	Farinacci	Composite brake rotors and clutches
5407035	Cole et al.	Composite disc brake and method of making
5261511	Libsch	Lightweight and high thermal conductivity brake rotor
4756392	McMurray	Stainless steel brake rotor for airplane disk brakes
UK 2268511	Cole et al.	Composite disk brake rotor with self-lubricating coating and method of making thereof
5618635	Newkirk	Macrocomposite bodies
5654107	Tanaka et al.	Wear-resistant aluminum alloy composite material
5514480	Takagi	Metal base composites
5223347	Lhymn	Creep-resistant composite alloys
5240782	Bose et al.	Fiber and whisker reinforced composites and method for making the same
5166004	Bose et al.	Fiber and whisker reinforced composites and method of making the same
5041340	Ushio et al.	Fiber reinforced light alloy member excellent in heat conductivity and sliding properties
6136374	Reuscher	Method and apparatus for coating vented brake rotors
5980651	De Azevedo et al.	Rotors and drum brake systems
5901818	Martino	Brake rotors with heat-resistant ceramic coatings
5620791	Dwivedi et al.	Brake rotors and methods for making the same
5526914	Dwivedi et al.	Brake rotors, clutch plates and like parts and methods for making the same
5028494	Tsujimura et al.	Brake disc material for railroad vehicle
6030913	Heine et al.	Fabrication of C/SiC composites
5372222	Rhee et al.	Lightweight and high thermal conductivity brake rotor
5224572	Smolen	Lightweight brake rotor with thin, heat-resistant ceramic coating

Targeting the wear resistance and weight saving aspects, a large number of patents on composite brakes have been filed. For aluminum and magnesium-based composites, reinforcements such as silicon carbide, silicon nitrate, boron nitrate, alumina, aluminum borate and potassium titanate have been considered. Most of these reinforcements promote the wear-resistant behaviours of the matrix alloys, and the low-density aluminum and magnesium alloys definitely offer weight savings.

In this study, the technical analysis was limited to the following three material types:

1. silicon carbide particulate reinforced aluminum (SiC/Al – commercial Duralcan materials),
2. CANMET modified alumina particulate reinforced aluminum (Al_2O_3/Al), and
3. a ceramic matrix composite (specifically carbon fibre reinforced silicon carbide matrix (C/SiC)).

The first two are metal matrix composites, while the third is a ceramic matrix composite. Of the three types of composites, a significant amount of data has been reported on SiC/Al, and very little on Al_2O_3/Al and C/SiC. Nonetheless, a preliminary evaluation is conducted on the viability of using these composites for brake rotors in heavy vehicles, based on both published data and preliminary modelling work performed at CANMET.

3.2 Test Methods and Standards for the Evaluation of Brake Rotor Performance

The process for brake material selection and verification is very rigorous and demanding. Initially, candidate materials are selected based on published information and material databases. Subsequently, the selected materials are evaluated for physical, chemical, corrosion and wear-resistance properties. Most of the test methods used [36,37,38] must comply with national or international standards. Finally, a systematic tribological evaluation of materials, in many cases both small-scale and full-scale tests, is conducted to complete the material selection and verification process. Figure 3-1 illustrates the flow of evaluation processes adopted by most brake manufacturers.

3.2.1 Small-Scale Tests

Small-scale tests are used to reduce the costs associated with the preliminary materials selection. These tests are not intended to replace full-scale testing, but are meant to perform initial materials screening. They are set up to provide a comparison between potential materials.

An ideal brake rotor material must provide a high and stable frictional performance with low wear and minimal noise and vibration. These characteristics must be maintained over a wide range of pressures and temperatures. For evaluating such characteristics, a sample list of standard tests used in the industry for the preliminary evaluation of brake rotor materials is included in Table 3-2.

Recently, a combination of small-scale tests and modelling approaches has been developed to provide a more precise evaluation of the frictional behaviour of materials over a typical range of operating conditions [39, 40].

3.2.2 Full-Scale Tests

Dynamometers are used mostly for controlled wear tests, basic effectiveness tests, initial fade recovery tests, and parking brake tests. No international standards exist for the specifications of dynamometer tests, as they are often used to simulate a specific vehicle, vehicle braking system, or operating condition (e.g., city bus, alpine descent, city driving).

In many instances small-scale tests are not conducted and full-scale tests on dynamometers are used instead for the initial stage of friction evaluation. This is because of the poor correlation between small-scale tests and dynamometer tests. Further, in many applications it is the minimization of judder and noise that are paramount, and this can only be determined reliably with full-scale tests.

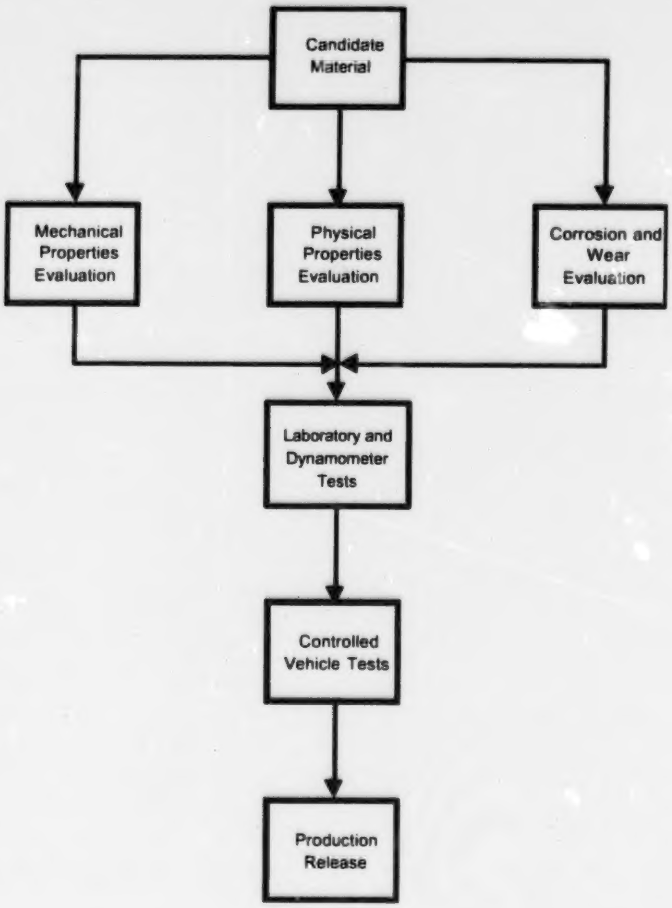
Candidate Material
Mechanical Properties Evaluation
Physical Properties Evaluation
Corrosion and Wear Evaluation
Laboratory and Dynamometer Tests
Controlled Vehicle Tests
Production Release

Figure 3-1 Procedures for Brake Rotor Material Selection and Verification

Table 3-2 A Sample List of North American Test Standards

North American Test Standard	Designation
Wear of Materials During Sliding – Using a Pin on Disc Machine	ASTM G99-95a
Friction Test – Using the Chase Machine	SAE - J661
Determination of Thermal Conductivity of Brake Linings by Guarded Hot-Plate Apparatus	ISO 7882
Evaluation of Effects of Heat on Dimensions and Form of Disc Brake Pads	ISO 6313
Determination of the Specific Gravity of Brake Linings	SAE J380
In-Service Brake Performance Test Procedure – Vehicles over 4500 kg (10 000 lb.) (1992)	J1250
Simulated Mountain-Brake Performance Test Procedure (1993)	J1247
Structural Testing of Passenger Car and Truck Disc Brakes (1998)	J1713
Brake Performance and Wear Test Code Commercial Vehicle Inertial Dynamometer (1993)	J2115
Dynamometer Effectiveness Characterisation Test for Passenger Car and Light Truck (1995)	J1652
Parking Brake Structural Integrity Test Procedure – Vehicles over 10,000 lb. (4500 kg) (1987)	J1476
Service Brake Structural Integrity Requirements – Vehicles over 10,000 lb. (4500 kg) (1993)	J1404
In-Service Brake Performance Test Procedure – Vehicles over 4500 kg (10,000 lb.) (1992)	J1250
Simulated Mountain-Brake Performance Test Procedure (1993)	J1247
Brake System Performance Requirements – Truck, Bus, and Combination of Vehicles	J992
Brake Power Rating Test Code – Commercial Vehicle Inertia Dynamometer (1991)	J971
Brake System Road Test Code – Passenger Car and Light-Duty Truck (1997)	J843
Brake Test Code – Inertia Dynamometer (1961)	J667
Automotive Brake Definitions and Nomenclature	J656
Automotive Grey Iron Castings (1996)	J431
Stopping Distance Test Procedure (1993)	J299
Service Brake Structural Integrity Test Procedure – Vehicles over 4500 kg (10,000 lb.) (1993)	J294
Service Brake Structural Integrity Test Procedure – Passenger Car	J229
Brake System Dynamometer Test Procedures – Passenger Car	J212
In-Service Brake Performance Test Procedure – Passenger Car and Light-Duty Truck (1997)	J201
Brake System Road Test Code – Passenger Car and Light-Duty Truck-Trailer Combination (1993)	J134
Brake System Road Test Code – Motorcycles (1987)	J108

Application tests on dynamometers are usually split into performance and endurance types. A performance type test is based on proving the brake's friction performance under dry, wet and overload conditions, as well as the component part's integrity at the extremes of the environment. An endurance type test is based on vehicle route simulations and is an accelerated method of evaluating disc and pad wear and hence service life under general operating conditions.

For automotive applications two specific braking manoeuvres are particularly onerous for the brake materials. These are "alpine descents" and "autobahn stops". Alpine descents involve descending a long steep slope while maintaining constant vehicle speed using the brakes. Autobahn stops involve retarding a vehicle from maximum speed to low speed or to a stop in a short period of time, often with a fully laden vehicle. Although these stops may only last a few seconds, extremely high peak temperatures may be generated at the friction interface.

It should be noted that even full brake dynamometers are difficult to correlate with on-road vehicle brake performance, unless careful instrumentation and test controls are used [41]. In many instances, extensive vehicle testing is performed in various geographical locations, over varying terrain and seasons, before brakes are released for commercial production. This is, of course, an expensive and time-consuming process.

The majority of the technical literature evaluating drum and rotor materials describes tests that are based on full-scale components. The geometries tested are generally based on the dimensions that have traditionally been used for cast iron. This approach allows a rapid preliminary evaluation of the capabilities of different materials. However, direct material substitution for cast iron in a drum or rotor component with geometry originally designed for cast iron may not take advantage of the thermal properties of the substituted material. With the exclusion of melting point, aluminum-based materials have significantly better thermal properties than cast iron. Direct substitution does not offer a true representation of what performance levels can be obtained from Al MMC materials if the rotor geometry is not optimized to take full advantage of these properties [42].

In view of the rigorous and demanding tests required in brake material selection and verification, it is unlikely that the test data available in the literature would provide any indication of the suitability of using composite materials for brake rotor application in heavy vehicles. Consequently, modelling work was conducted to provide some perspectives on temperature and stress profiles experienced by brake rotors in heavy vehicles.

3.3 Brake Drum and Rotor Material Properties

In the selection of brake materials, weight is not the only concern. There are many physical, mechanical, thermal and corrosion property requirements that must be taken into consideration. Each of these properties will influence the performance characteristics of a brake rotor. To gain a better understanding regarding the selection of an MMC for brake rotor application, each of the physical, mechanical, thermal and chemical properties that is linked to the performance characteristics of a brake rotor will be discussed.

It should be understood that successful material substitution for cast iron can only be achieved by maximizing the benefits derived from the selection of the most appropriate combinations of properties available. For this reason, it is essential that a designer is aware of the key material properties corresponding to the functional performance of brake rotors, and makes the material selection or fabricates new materials on that basis. From the review of previous literature, it appears that the key properties and functional performance of brake rotors are those listed in Table 3-3. Each of these properties is discussed in this section.

Table 3-3 Brake Rotor Materials and Functional Properties

Material Property	Functional Properties
1. Density	Weight Reduction
2. Coefficient of Thermal Expansion	Dimensional Stability, Thermal Fatigue Resistance
3. Thermal Conductivity	Heat Dissipation
4. Specific Heat	Heat Absorption
5. Thermal Shock Resistance	Reliability in Service
6. Thermal Fatigue	Reliability in Service
7. Elevated Temperature Properties	Friction Surface Integrity, Maximum Service Temperatures
8. Strength	Dimensional Stability
9. Elastic Modulus	Dimensional Stability
10. Toughness	Reliability in Service
11. Ductility	Reliability in Service
12. Wear Resistance	Braking Response and Service Life
13. Damping Capacity	Noise, Vibration and Harshness
14. Corrosion Resistance	Service Life and Ride Comfort
15. Machinability	Cost
16. Processability	Cost and Reproducibility

3.3.1 Density

The main advantage of replacing cast iron with aluminum is the saving in weight. Cast iron has a density of 7.25 g/cc, while the density of aluminum is only 2.7 g/cc. Most ceramic reinforcements for MMCs have a slightly higher density that aluminum, but are in the range of 2.5 to 4.5 g/cc. For example, the density of silicon carbide particulate reinforced aluminum composite with 20 volume percent reinforcement would have a density of 2.8 g/cc. Consequently, from a weight reduction standpoint, there is a definite advantage in replacing cast iron with aluminum based MMCs.

As for C/SiC, the density of carbon fibre is 1.8 g/cc, while silicon carbide has a density of 3.2 g/cc. The combination of the two gives the C/SiC composite a density of 2.4 g/cc, which is substantially lower than that of cast iron.

3.3.2 Coefficient of Thermal Expansion (CTE)

CTE is an important property to be considered in the selection of a brake rotor material. A material with a high CTE means the material will expand substantially at elevated temperatures. For brake rotor application, a material with a high CTE is undesirable, since it may lead to distortion and thermal fatigue due to the development of thermal stresses at elevated temperatures. Unfortunately, CTE values for most metals are normally high. The CTE for cast iron is 12.1 ppm/K and the CTE for aluminum is 23.8 ppm/K. Fortunately, CTEs for ceramics, such as silicon carbide and alumina, are lower than metals. For example, SiC has a CTE of 4.3 ppm/K.

Consequently, if one were to embed ceramic particulate in aluminum to form an MMC, it would have a CTE value lower than that of unreinforced aluminum [43]. In the case of C/SiC, since both carbon and silicon carbide have extremely low CTE values, the resulting C/SiC composite has an extremely low CTE value of 3 ppm/K.

3.3.3 Thermal Conductivity

Thermal conductivity describes the property for heat dissipation in a material. Aluminum has a much higher thermal conductivity value (200 W/m· K) than cast iron (58 W/m· K). As for aluminum MMC with silicon carbide particulate reinforcement, a thermal conductivity of around 180 W/m.K can be expected. With the higher thermal conductivity of MMC compared to cast iron, one of the key advantages of replacing cast iron with an aluminum MMC is the reduction of thermal stress gradients on the brake rotor caused by localized "hot spots" on the surface [44]. Another interesting property is that the thermal conductivity of aluminum increases with temperature. That is, aluminum-based composites with silicon carbide reinforcement will have over three times the thermal conductivity of cast iron at room temperature and almost five times at around 400°C.

The thermal conductivity of C/SiC is not high, and a value of 40 W/mK was reported. However, if very high thermally conductive carbon fibres are used as reinforcement, it is possible to improve the thermal conductivity of C/SiC.

3.3.4 Specific Heat

This property defines the amount of heat absorbed to raise the temperature of a material by 1°C per unit weight. It also relates to the amount of heat that a material is capable of absorbing. The specific heat for a typical cast iron is 5 MJ/K· m^3. For SiC/Al MMC it is 3.2 MJ/K· m^3, while for C/SiC it is 3.8 MJ/K· m^3. If based on the specific weight, both SiC/Al and C/SiC will absorb more heat than cast iron for each degree rise in temperature.

For brake rotor application, it is desirable to have brake rotor materials with high specific heat. The reason for using a high specific heat material for a brake rotor is that it doesn't heat up as easily during braking, and thus experiences less thermal stress.

3.3.5 Thermal Shock Resistance

Thermal shock resistance is the ability of a material to withstand thermal stresses brought about by sudden and severe changes in the temperature at the surface of a solid body. The thermal shock resistance of a material can be estimated in terms of a thermal shock parameter or Eichelberg factor (K), expressed by the following equation:

Thermal shock resistance $K = \sigma S / E \alpha$

Where σ is the thermal conductivity,

S is the tensile strength of the material,

E is Young's modulus, and

α is the coefficient of linear expansion.

By plugging in property values for cast iron, SiC/Al and C/SiC, the K data for these materials are 6,800 W/m, 40,000 W/m and 46,000 W/m, respectively. It is apparent that the thermal shock resistance values of SiC/Al and C/SiC are an order of magnitude greater than those of cast iron.

3.3.6 Thermal Fatigue

Thermal fatigue occurs when the heating and cooling cycles cause accumulation of residual stresses in a material. The amount of residual stresses accumulated depends on the magnitude of thermal stresses imposed on the material and its material properties. Thermal stresses are related to Young's modulus E, thermal expansion coefficient α, and temperature difference δT, by the following equation:

$$\text{Thermal Stresses} = E \cdot \alpha \cdot \delta T$$

As the thermal cycles are repeated, the accumulating residual stresses may exceed the endurance limit of the material, thus causing eventual failure. To minimize thermal fatigue, a material should have a low modulus and a low coefficient of thermal expansion. Both properties are significantly lower in the composites than cast iron.

3.3.7 Elevated Temperature Properties

In addition to the thermal properties mentioned above, it is important that a brake rotor material has stable properties at elevated temperatures [45,46] or, more precisely, at maximum service temperatures. Therefore, a parameter called maximum operating temperature (MOT), is introduced to indicate the maximum operable temperature of the material. For example, the melting point of cast iron is 1300°C, whereas the melting temperature of SiC/Al is only 550°C. However, when one takes into consideration the lack of resistance to oxidation in cast iron at high temperature, and the fact that it does not retain sufficient high temperature strength, the serviceable temperature is significantly lower than the melting temperature. In summary, it is important to use the MOT to judge whether the selected material is suitable for use in the service temperature range. For SiC/Al and C/SiC, the MOTs are between 450 to 490°C and 1350°C, respectively, and 700 to 800°C for cast iron.

3.3.8 Strength

Strength is the most basic property of a material. For brake rotor application, a high strength material is required. The tensile strength of a cast iron is 200 MPa, and that of a typical aluminum MMC (silicon carbide particulate reinforced aluminum) is 330 MPa. The high strength property in a brake rotor is required to carry clamping loads and braking torque, and to resist thermal stresses.

It is well known that the strength of both aluminum MMC and cast iron decreases with the increase of temperature. However, cast iron definitely has an advantage, since the tensile and compressive strengths of cast iron are more than twice that of aluminum MMCs at about 540°C.

3.3.9 Elastic Modulus

Elastic modulus relates to the stiffness of the material. For cast iron, the modulus is 200 GPa and for aluminum alloys, the modulus is only 69 GPa. Therefore, aluminum by itself is too malleable to be used for rotor application. Reinforcing aluminum with stiff ceramic particles increases the stiffness of the material. The extent of modulus increase is dependent on the volume fraction of the reinforcement. In general, MMC stiffness increases linearly with increasing particle content. If the composite is properly fabricated, it is possible to fabricate aluminum composites (with particle reinforcement) with stiffness values roughly twice those of the base materials.

A brake rotor material should have a suitable modulus. A material with high modulus would lower the thermal shock resistance, while a low modulus material might be more prone to mechanical deformation.

3.3.10 Toughness

Toughness is the ability of the material to absorb energy during plastic deformation. This is an important property for service reliability as well as during brake rotor manufacturing. Toughness in composites is very dependent on the fabrication process. In a composite, the fabrication process conditions significantly influence the interfacial properties between the reinforcement and the matrix alloy in a composite. It is these interfacial properties that determine the toughness of a composite.

3.3.11 Ductility

Generally, MMCs have a lower ductility than their unreinforced counterparts. Such is the case for SiC/Al, the low ductility of which causes it to have a higher propensity for crack formation and growth than iron-based materials. This not only influences the selection of processing and machining methods but also increases the propensity of the material to suffer from thermally induced fatigue cracks. Cyclic thermal loading of a brake disc results in alternating tensile and compressive loads that can produce radial cracks propagating toward the inner edge of the disc. Therefore, for brake rotors, high-strength aluminum alloys are selected for the base material, including alloys A356 [46] and A359 [47].

3.3.12 Wear Resistance

Under both sliding and abrasion wear, the wear rate of MMCs decreases with an increase in volume fraction. Abrasive wear rates can be reduced by 55 to 90 percent from the wear rate of an unreinforced aluminum alloy, depending on the volume fraction of the reinforcement.

When a SiC/Al composite with 20 volume percent reinforcement is slid at low speeds against organic friction pads developed for cast iron rotors, a coefficient of friction value lower than that found for cast iron is observed. At higher sliding speeds, however, the steady-state friction value for MMCs against the organic pad exceeds that of cast iron.

In essence, in the case of SiC/Al, the aluminum asperities (nodules) are worn down until the discrete particles of hard silicon make contact with the friction lining. The level of SiC particles is raised slightly above that of the aluminum matrix material [47] and it is these hard reinforcing particles that carry the majority of the applied load and constrain the matrix material from excessive deformation, thus reducing the wear and friction.

At low loads, the wear rate of the SiC/Al with low iron brake pad (developed specifically for MMCs) is low. However, at higher loads and sliding speeds, SiC/Al exhibits a high wear rate.

3.3.13 Damping Capacity

Material damping is a manifestation of internal energy dissipation in a material during dynamic deformation. In other words, damping capacity is defined as the energy absorbed during a cycle of vibration. Damping is closely linked to the suppression of noise and vibration. High damping capacity materials help to meet requirements for increased quietness and durability of structural materials under vibratory loading.

From the material's standpoint, damping capacity is directly related to the internal friction of the material, and it is the extent to which the vibration amplitude is reduced. For a metallic material, most of the damping is provided by the matrix alloy. In the case of MMCs, most ceramic reinforcements have a lower damping capacity than metals. However, the combination of metal and ceramic might provide a higher damping capacity material. There are other significant factors which contribute to the increase in damping capacity of a composite. These factors are:

- increased dislocation density in a composite as a result of the huge differences in the coefficients of thermal expansion between ceramic particles and the matrix;
- increased particle-to-matrix interfaces; and
- increased grain boundaries due to the fine grain size of a composite.

3.3.14 Corrosion Resistance

The corrosion resistance of aluminum matrix composites is much better than that of cast iron when exposed to road conditions of high humidity, severe cold and the presence of salt. Moreover, C/SiC provides even better corrosion resistance than the aluminum-based composites.

3.3.15 Machinability

Because of the abrasive nature of ceramic reinforcements such as silicon carbide, the machinability of SiC/Al is much poorer than cast iron. Often diamond tooling is required for the machining of SiC/Al composite brake rotors. This increases the cost of machining if SiC/Al brake rotors are to be used. One alternative for reducing the cost of machining composite brake rotors is using alumina particles instead of silicon carbide particles. The fine nature of alumina particles is expected to improve the machining characteristics of the composite.

The other alternative is to provide reinforcements only at the wear surfaces of the brake rotor and not in the hub region. In this manner, when the major portion of the machining is required at the hub region, no extra costs involving diamond tooling are incurred.

3.3.16 Processibility

The manufacturing process for of all composite materials is more complicated than that for cast iron. This is especially the case for C/SiC composites. Process descriptions and analysis are provided in chapter 4 of this report.

3.4 Brake Material Performance Requirements

In improving the performance of brakes, the two basic approaches are to use a better material or adopt a new design. Regardless of the approach taken, it is important to have a thorough understanding of the material properties and functional performance relationship, as well as service conditions, to come up with a suitable solution.

For example, there are two distinct approaches in the design of brake rotors for family-sized vehicles, and larger or high-powered vehicles [48]. For family-sized vehicles with smaller diameter rotors, it is important that the rotors have high strength and that they resist thermal cracking and distortion at high operating temperatures. For larger or high-powered vehicles, rotors can be thicker and have a larger diameter, with lower strength and higher thermal conductivity. Regardless of whether the design is for small or large vehicles, many of the performance characteristics required in those rotors are similar. In this section, emphasis is placed on the performance characteristics (reported in the literature) of brake rotors rather than on the rotor materials.

3.4.1 Tribological Characteristics

Wear characteristics of a brake rotor are sensitive to temperature, rubbing speed, pressure, and most importantly, the material composition. The overall brake design and contact geometry, the removal of wear debris, and the environment also have major effects. In the case of SiC/Al, the fragmentation of SiC particulate reinforcements would lead to the formation of third body wear, leading to a more severe wear. All of these factors must be considered in a complete tribological characterization of the friction material pair. Any friction peaking fade (an increase in friction occurring during or after high temperature operation) or fade (a temporary reduction in braking effectiveness due to loss of friction between the braking surfaces resulting from heat) effects must also be characterized.

In essence, from the wear resistance standpoint, a suitable brake rotor material must have an acceptable loss in mass or change in volume during service, and a high wear-resistant material is preferred. The friction material against the rotor must be as weak as possible. If the attack of the friction material against the rotor is too severe, then this can lead to frequent machining or replacement.

3.4.2 Corrosion Characteristics

Corrosion is a major issue for brake rotors used in Eastern Canada because of the severe winter conditions and the use of salt in deicing roads. Corrosion in brake rotors leads to a loss of cooling efficiency, braking surface damage, and degradation in structural integrity. It is extremely desirable to have brake rotor materials that demonstrate higher corrosion resistance than cast iron.

3.4.3 Thermal Behaviour

Thermal behaviour is the most important property in brake rotor materials. Quite a number of papers have recently been published in this area [49,50,51,52,53]. The three mechanisms by which heat is transferred away from a brake rotor's surface are conduction, convection and radiation.

At high temperatures cast iron materials dissipate a significant amount of heat by radiation. However, for Al MMC rotor materials radiation is not a significant heat dissipation mechanism because of their low operating temperature. Thermal conduction and forced convection are the dominant heat transfer mechanisms for Al MMC rotors over the operating temperature range. Thus, the rotor surface, rotor volume and the cooling airflow around the rotor all have a very significant effect on the Al MMC rotor's thermal performance [42].

The thermophysical properties of Al MMCs have different effects on the brake operating temperature depending on the braking duty. For light braking, the peak rotor temperature is determined predominantly by the product of the specific heat and the density of the material. At intermediate and high braking duties, the material's thermal conductivity is the dominant thermal property in determining the rotor temperature. Therefore, during light braking Al MMC rotors will run at a higher temperature than cast iron rotors, whereas during heavy braking Al MMC rotors will run cooler than cast iron rotors. This lower operating temperature provides the opportunity for superior frictional stability under heavy braking duties [43]. Lower operating temperature is desirable for the benefits of improving lining life and reducing rotor wear and squeal.

There are various ways to lower the operating temperature of a brake rotor:

- The rotor is designed to maximize the cooling effect. Improvements to vehicle aerodynamics reduce the airflow to the inside of the wheels. As cast iron exhibits very limited sensitivity to air flow cooling, operating temperatures may be reduced only very slightly as a result of large increases in cooling airflow. Conversely, the thermal performance of an Al MMC rotor is highly dependent on a good supply of cooling air [52]. Vented disc designs are most commonly adopted for use with Al MMCs because of the improved airflow around the disc. It is often found that relatively minor redesigns of the rotor venting geometry can yield significant improvements in vented cooling airflow. These improvements in airflow allow more rapid convection of heat from the material and thus lower the operating temperature of the disc [42,54,55].

- In addition to design alteration, changes in the dimensions of a brake rotor have a significant impact on its operating temperature. Based on a Rover 200 vented front brake disc geometry, the thickness of the disc has a significant effect on the predicted peak alpine descent temperature of the disc. For a given disc material conductivity, the predicted peak alpine descent temperature is reduced as cheek thickness is increased. A similar reduction in the predicted peak alpine descent temperature of the disc may be achieved by increasing the disc vent width. Finite element modelling has shown that disc cheek thickness has a dramatic effect on the predicted peak autobahn stop temperature. For example, increasing the cheek thickness from 7 to 9 mm may reduce the predicted autobahn temperature by over 100°C. Selection of discs with thick cheeks and wide vents can be expected to place the MOT of SiC/Al safely above the maximum predicted disc temperature for alpine and autobahn braking manoeuvres [56].

- Further improvements to disc operating temperature can be achieved by optimizing the rotor volume and the ratio of rotor volume to surface area. The low density of Al MMCs allows the disc diameter to be increased to the boundaries of the available wheel space while maintaining significant weight savings. The increased torque arm at which the frictional force now operates, allows reduced friction forces to be used while still providing the same braking torque. Thus, caliper size, weight and cost can be minimized. Quantitative estimates of the extent to which this may be possible include a disc diameter increase of 15 percent and a piston area reduction of almost 20 percent.

- Rotor operating temperature is highly dependent on the thermal conductivity of the rotor materials. Finite element models used for optimizing material selection have shown that a good correlation exists between a material's thermal conductivity and the peak operating temperature during an alpine descent. For example, materials with a thermal conductivity of 100 W/(m.K) may reach a peak temperature of approximately 580°C, whereas materials with a thermal conductivity of 200 W/(m.K) may reach a peak temperature of around only 510°C.

To assess the suitability of a brake rotor material, the MOT parameter is often used. MOT is the maximum operating temperature which a rotor can withstand prior to experiencing some undesirable surface melting. The predominant failure mode of SiC/Al brake material is when the glaze (typically forms on the rubbing surfaces of the rotor) breaks down at MOT and scuffing begins. Failure is usually considered to have occurred when the frictional force has exceeded the shear strength of the rotor material. This breakdown leads to excessive noise, sparks, dust and rapid wear. There is no industrial standard on the required MOT; however, the preferred temperature is at least above 482 °C.

The MOT value has been used as a quantifiable value for comparing different materials. Investigations by major manufacturers [57,58,59] concentrating on aluminum-based composites have found that a composite with a low MOT of 450°C would give rise to failure of the rotor surface by excessive scoring.

This can be attributed to several factors, such as inadequate cooling of the disc, too high a braking duty, and poor material properties.

For aluminum composite materials, the maximum operating temperatures can be raised by:

- using a high solidus temperature aluminum alloy, or
- increasing the volume fraction of reinforcement.

For example, the MOT of an aluminum 360 alloy reinforced with 20 volume percent of SiC particulates (typical of the current Duralcan product) is 449°C [60]. The MOT of the aluminum 360 alloy reinforced with SiC particulates becomes 482°C when the volume fraction of SiC particulate reinforcement is increased from 20 to 30 volume percent. There are, however, mixing concentration limits for the manufacturing process that make increasing the SiC concentration difficult. Because SiC is very expensive, any increase in its percentage has major effects on parts cost.

Two approaches to increasing the relative heat resistance of the aluminum are to use the CANMET matrix design or to preferentially remove aluminum from the material surface through chemical etching, thereby artificially increasing the concentration of SiC at the wear surface. This latter approach has been used by Volvo with some success in getting a more stable friction layer to form and increasing the temperature tolerance of the MCC brake part [6]. Volvo's test indicated that the MMC material could withstand temperatures in excess of 450°C with no wear or structural problems.

The other factor that influences the MOT of aluminum composites is the solidus temperature of the aluminum alloy. It has been found that by replacing aluminum 360 alloy with aluminum 1 percent magnesium alloy, the MOT was increased to 580°C [47]. In summary, the two primary factors influencing the MOT of a material are the volume fraction of reinforcement and the solidus temperature of the matrix alloy.

The relationship between MOT, solidus temperature of the matrix and the volume fraction of SiC reinforcement found by Dwivedi [60] is:

$$\text{MOT} = (\text{Solidus Temperature})\,(0.668 + 0.00564\,(\text{Vol. percent of SiC}))$$

Thermal fatigue exists when a heating and cooling cycle causes the accumulation of residual stresses. The magnitude of the thermally induced stresses is largely a function of thermal expansion, thermal conductivity, specific heat, modulus of elasticity and chemical compositions. In the event that differential thermal expansion between the inner and outer brake rotor surfaces exists, and the temperature cycle is of significant magnitude, the stresses will exceed the yield strength of the material, leading to fatigue-type failure.

It is crucial for a brake rotor material to be able to resist high thermal stresses (resulting from repeated thermal cycling) experienced during use. The large amounts of heat generated during braking must be absorbed and dissipated quickly. Therefore, the thermal properties of the materials must be optimized to accommodate the frictional heat generated at the rubbing interface.

3.4.4 Mechanical Behaviour

The dimensional stability of a brake rotor is linked to the stiffness of the brake rotor. In most cases, the stiffness of brake rotor materials decreases with an increase in temperature. Consequently, it is desirable that the brake rotor's stiffness not be easily lowered at high temperature during braking. That is, the rotor should remain dimensionally stable and not generate rotor thickness variation (RTV) after repeated braking.

There has been no apparent solution found for the unacceptable RTV problem for cast iron. However, the lower operating temperatures, better dimensional stability and higher wear resistance of SiC/Al rotors result in significantly less susceptibility to this phenomenon. Final RTV levels of SiC/Al are well below the normal acceptance level of 0.01 mm [61].

3.4.5 Physical Behaviour

Weight reduction in a vehicle is a very important factor. First, a heavy brake rotor contributes to the weight of a vehicle, causing a reduction in fuel efficiency and an increase in greenhouse gas emission. Second, being part of the unsprung weight of a vehicle, a heavier brake rotor tends to cause more NVH. Third, as part of the moving components of a vehicle, a heavy brake rotor requires additional energy for its motion and causes greater vibration.

There are only two means of reducing the weight of a brake rotor. The first is to change the brake rotor material, while the second is to develop a new design. Considering that the design of conventional brake rotors is mostly optimized, the most likely means of reducing the weight of a brake rotor is to use a lighter weight material. In terms of weight, MMC rotors can provide up to a 60 percent reduction when compared to cast iron [62].

3.4.6 Acoustic Behaviour

Reduction of brake noise and vibration contributes to improved vehicle quality passenger comfort and brake system reliability. A high level of damping in brake components contributes to suppression of brake noise and vibration.

The formation and destruction of a transfer film, as well as its coverage on a brake rotor, was found to be a controlling factor on the wear behaviour of the brake rotor [63]. Links between the presence of friction films and noise and vibration characteristics have also been examined [64]. It has been speculated that brake noise is caused by transfer film destruction. Being free from abnormal noises (usually called brake noise, or squeal) and abnormal vibrations (called brake judder) is of great importance to vehicle manufacturers. Their requirements are to match consumer expectations for maximum comfort and durability. Noise and vibration problems are usually attributable to the friction material and play a significant role in the selection of new friction pairs. The thought behind the transfer film destruction hypothesis relates to the unstable friction coefficient caused by the thermal destruction of the film, thus resulting in vibration and eventually noise from the brake system. Brake noise is usually associated with steady elevated brake temperatures at which the friction films on the rotor and pads are well developed and the coefficient of friction remains steady.

Aluminum-based composite brake rotors have consistently been shown to have low or no noise, especially with respect to squeals and creep groan.

3.5 Property and Performance Summary

The key property and performance data (mentioned in sections 3.3 and 3.4) for brake rotors made with cast iron, SiC/Al and C/SiC are summarized in Table 3-4. From the foregoing review, the following conclusions can be drawn:

- SiC/Al has the best thermal and mechanical properties. However, because of the low melting aluminum matrix alloy, the MOT is only 450°C. Therefore, the current commercial SiC/Al is not really suitable for brake application if the service environment demands a higher MOT.

- A complete set of property data on CANMET modified Al_2O_3/Al is not yet available, but the results of preliminary equivalent tests indicated a higher MOT (around 600°C) is achievable. C/SiC has the highest MOT (1350°C) and is suitable for heavy vehicle rotors.

Table 3-4 Properties of Brake Rotor Materials

Property	Cast Iron	SiC/Al	C/SiC
Density (gm/cc)	7.3	2.7	2.4
Thermal Conductivity (W/m.K)	55	240	40
Coefficient of Thermal Expansion (10^{-6} K)	12	20	3
Heat Capacity (MJ/K. m^3)	5.0	3.2	3.8
Thermal Shock Resistance (W/m)	6,800	44,000	46,000
Maximum Operating Temperature (°C)	700	450	1350
Bending Strength (MPa)	200	300	80
Young's Modulus (GPa)	110	100	35
Bending Strength (MPa)	200	300	80
Life Expectancy (km)	60,000	250,000	300,000
Corrosion Resistance	Poor	Good	Excellent
Weight Saving (%)	0	64%	68%
Front Brake Application	Yes	No	Yes
Elongation (%)	0.5	0.45	0.3

3.6 Preliminary Assessment for Composite Brake Materials

The two main means of qualifying brake rotor materials are through:

1. extensive small- and full-scale tests, or
2. computer modelling with experimental data verification.

Both approaches are extremely costly and time consuming. Therefore, all the test data from the small- and full-scale tests are often proprietary. As for computer modelling, the established models by individual companies are also proprietary, and no algorithm is made available. Consequently, it is very unlikely that any organization with in-house material qualification capability is willing to provide a test model for evaluating composite materials. In fact, efforts made by the project team in acquiring such information proved to be fruitless.

Considering the situation, CANMET decided to perform its own modelling work to provide some estimation on the functional performance of composite brake rotors. In this model, special attention was focussed on the thermal and stress profiles of brake rotors during service. The modelling work consisted of three steps:

1. A model was developed using finite element methods (FEM).
2. The model was compared with the model developed by Roberts and Day [65]. The purpose of this second step was to provide a reality check on the model, since the Roberts and Day model was supported by field data.
3. The model was extended to predict the thermal and stress profiles of both disc and drum composite brakes used in a heavy vehicle.

To provide a realistic scenario of the service conditions of heavy vehicle brake rotors, the drum brake rotor design used in a 1995 Freightliner FLD-120 truck with a sleeper towing a 45 ft. (13.7 m) box trailer was modelled. The reasons for this choice were:

- The availability of a paper that provided similar modelling work using a commercial software, thus providing some useful raw data and information for reference purposes;

- The fact that a 48 ton truck is a good size heavy vehicle (in the modelling work, both disc and drum brake rotors made with cast iron, C/SiC and SiC/Al were modelled).

3.6.1 Thermal and Strength Analyses of Composite Drum Brake Rotors

The work/energy transition and heat transfer of three heavy-duty air brakes (mounted on a 1995 Freightliner FLD-120 truck), whose drums were made of cast iron, C/SiC, and SiC/Al, were modelled during an emergency stop following a downhill descent and analysed by FEM. The calculated temperatures of the lining and cast iron drum were compared with those calculated by the commercial Human-Vehicle-Environment program provided by Roberts and Day [65]. Stress analysis was also conducted for the drum during the emergency stops.

Thermal analysis

The typical size of a drum and linings of a heavy-duty air brake were used having an arc length of the two linings of 130° each. The thermal properties of three different drums made of cast iron, C/SiC and SiC/Al, and a lining are listed in Table 3-5. As in the Roberts and Day paper, the initial temperature of the drum and linings was chosen as 38°C (100°F).

Table 3-5 Thermal Properties of Drums and Lining

Property	Drum/Rotor Material or Lining			
	Cast Iron	C/SiC	SiC/Al	Lining
Density (g/cm^3)	7.3	2.4	2.7	1.959
Thermal conductivity (W/mK)	55	40	240	1.139
Specific heat (MJ/m^{3}K)	5.0	3.8	3.2	2.047

The brake torque for a 1995 Freightliner FLD-120 with a sleeper towing a 45 ft. (13.7 m) box trailer (total weight = 73,418 lb. or 33,302 kg) when descending a straight 6 percent downhill followed by an emergency stop was estimated by Roberts and Day. They estimated a torque of 3125 in.-lb. (353 N· m) when time t=0 to 150 seconds as the truck descended the downhill and torque of 97,200 in.-lb. (10,984 N· m) during time t=150 to 156 seconds when the truck made an emergency stop. The speed of the truck was assumed to be 65 mph (105 km/h) when descending downhill and changed to 0 during the emergency stop, which lasted six seconds.

During braking, the vehicle's kinetic energy is converted into heat energy at the drum/lining interfaces. This input of heat energy into the brake assembly is then dissipated to the atmosphere through both conduction and convection. In the mathematical model, a two-dimensional axisymmetric finite element model was used to simulate the drum/lining assembly, with the finite element mesh used in the calculation. The finite element mesh consists of three parts:

1. A solid line that outlines the exterior of the drum boundary represents convection elements. This was used to model the heat dissipation from drum to air.

2. A layer of drum elements inside the convection elements that have the thermal properties of the drum.

3. Another layer (the interior of the drum) that represents elements with the thermal properties of a lining.

Heat was generated in a tiny layer between drum and lining elements second by second. Finite element code ADINA was used by CANMET to perform the simulation and calculation in this work. Figure 3-2 shows the temperature profile just after the emergency stop.

The modelling indicates that:

- The temperature in the lining is lower than that in the drum because of its poor conductivity; therefore, less heat is transferred to it.
- The top part of the drum has a higher temperature than the lower part, where heat was being transferred to the bottom.

Figure 3-3 shows the temperatures of brake drums of these three materials. It can be seen that the temperatures in the drum are higher for the C/SiC drum than for cast iron and SiC/Al.

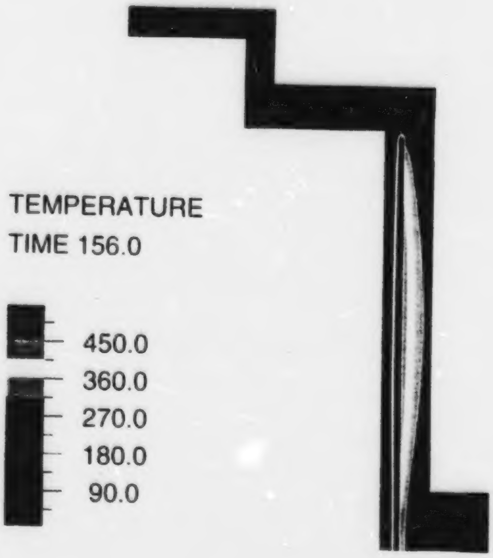

Figure 3-2 Drum Temperature Profile

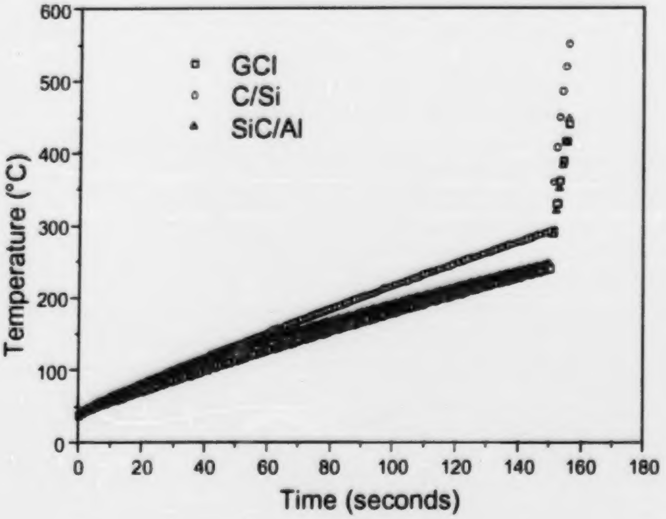

Figure 3-3 Temperature Histories of Drums Made of Three Different Materials

Stress Analysis

Figure 3-4 shows the various applied forces and stresses acting on the drum during braking. The major applied forces are the torque given by the wheel/ground, the pressure given by the lining/spring and the shear stress given by the lining through friction. The balance between the torque and the friction shear stress was modelled and the hoop stress calculated.

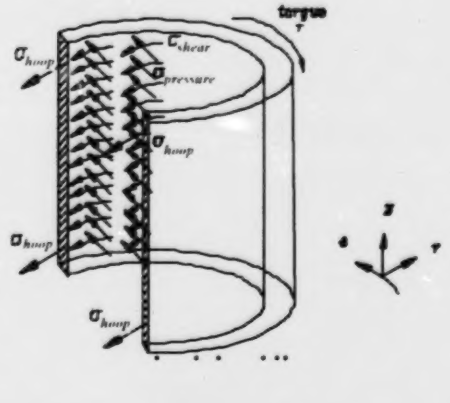

Figure 3-4 Forces and Stresses in a Drum Brake

For the torque estimated during the emergency stop, the calculated shear stress, the pressure, the hoop stress and the related effective stress were 0.3, 0.8, 10.6 and 12.96 MPa, respectively. To verify the analysis, two-dimensional axisymmetric finite element calculation was conducted for the drum.

The calculated maximum values of the local hoop stress and effective stress were 13.1 and 15.7 MPa, respectively. These are a little bit larger than those given by the above analysis. The reason for this difference is that the results of the finite element analysis are the maximum values and those given by the above analysis are average values. It should be noted that no stress concentration was observed at the bottom of the drum and the same maximum stress was obtained by using different radii between the well and the bottom of the drum. The calculated stresses (about 15 MPa) are well below the strengths of the three materials even when the thermal deterioration of strength is considered.

It was also estimated that the hoop stress and effective stress of the drum created by the largest torque during the emergency stop would be much lower than the bending strengths of all three materials.

3.6.2 Thermal and Strength Analyses of Composite Disc

Finite element analyses were conducted for two disc brakes: one for small to medium cars and the other for heavy-duty trucks. The analysis for the first gave the temperature prediction for a road test, which consists of 13 braking events during a drive of 67 seconds. By comparing the mode results with the available measured temperatures, it was shown that the calculated temperature was reasonably good. Using a similar model, the temperature profiles of a brake for heavy-duty trucks were modelled. Subsequently, the effects of using different brake materials (such as cast iron, C/SiC and SiC/Al) on the temperature profiles were estimated. As well, stress analysis for the disc brake was conducted.

Thermal Calculation

A finite element mesh was used for modelling the brake rotor for small cars with the rate of heat energy input estimated from the kinetic energy transfer rate measured during a road test [66]. It should be pointed out that only part of the heat transferred from the total kinetic energy (86 percent in the reference) was used as heat input for the disc since only the disc was modelled. The other part of the energy (14 percent) was dissipated through the connection of discs and car body. By using this heat-input rate, the temperatures at each stop (braking) were calculated and compared to the empirical data. A high correlation was shown.

A model similar to that used for the drum brake analysis was used to calculate the temperature of a heavy-duty air brake during a downhill descent followed by an emergency stop. All the thermal conditions, thermal properties and thermal equations used were the same as for the drum analysis but with a different FEM form.

The rate of heat transferred from kinetic energy was assumed to be exactly the same as in the drum model: 47.1 kJ/s during the downhill descent (0 to 150 s) and 732.6 kJ/s during the emergency stop (150 to 156 s). As only the disc was modelled, 86 percent of the heat transferred from the kinetic energy was used as heat input. The temperature vs. time curves for discs made of three different materials – cast iron, C/SiC and SiC/Al – are shown in Figure 3-5.

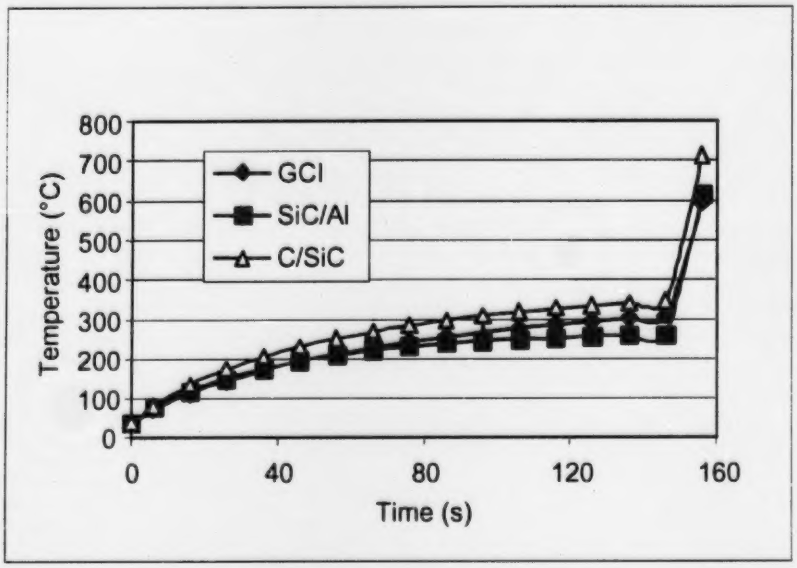

Figure 3-5 Disc Rotor Temperature History

From the plot, it can be seen that the disc made of C/SiC has the highest temperature. Of discs made of Grey Cast Iron (GCI) and SiC/Al, GCI has a higher temperature than SiC/Al for slow braking (0 to 150 seconds). But for emergency braking (150 to 156 seconds), the temperature of the SiC/Al increases more than that of GCI. The reason for this difference is that for slow braking, there is enough time for heat transfer, so the value of conductivity plays a very important role. SiC/Al has very high conductivity (240 compared with 55 for GCI), so the heat can be transferred to the surface easily. For the emergency stop, there is not enough time for heat transfer, so the value of specific heat plays a more important role than conductivity. SiC/Al has a lower value of specific heat (3.2 compared with 5.5) than GCI, so the temperature of SiC/Al increases more. As the stop and go test is a kind of slow braking, it is reasonable to conclude that SiC/Al maintains a lower temperature than GCI.

Stress Calculation

Maximum stress in the rotor model was assessed using FEM analysis assuming a 60 degree pad. The calculated maximum shear stress was estimated at 1.81 MPa, which is within the material's physical capacity.

3.7 Volvo MMC Developments

During the course of this study, Volvo Technology Development Corp. published information [6] on improving thermal performance of MMC materials through the use of slightly wider particle sizes for the 20 percent SiC in the aluminum ($AlSi_9Mg$ with a particle size range from 13 to 23 μm) and through chemical etching of the wear surface of the finished casting. The brake chosen for the development work was the front brake for a Volvo S40 with 15-inch wheels. The size was constrained to allow direct substitution with the existing cast iron disc rotors. The work is judged to be very significant to the future use of MMC in brakes.

The standard cast iron front discs are ventilated with a diameter of 281 mm, thickness of 24 mm, and weight 6.3 kg. They were replaced by solid MMC discs of the same diameter, with the thickness increased to 26 mm to enhance the thermal capacity. Disc weight was reduced to 3.9 kg. The standard rear discs are solid with diameter 260 mm, thickness 10 mm, and weight 3.4 kg. These were replaced by solid MMC discs with diameter 2 mm, thickness 12 mm, and weight 1.6 kg.

The relatively wide particle size distribution was chosen to reduce the material costs while larger particles were used to generally improve the particle pull-out strength from a matrix. The maximum particle size was chosen to avoid excessive settling of particles during melting and casting, and to prevent damage to the diamond machining tooling. A direct squeeze casting process was used for a number of reasons:

- High degree of material utilization (important because of relatively high raw material costs);
- Near net shape production (reduces the amount of machining);
- Lowest possible level of porosity (close to surface porosity negatively affects the temperature stability of the friction surface).

The Volvo design also avoided the use of heat treatment, thus eliminating a normal production process cost.

A prerequisite for proper function of an MMC-based brake is a stable friction transfer layer (the layer between the drum or disc and the brake pad). Volvo (and others) have found that different pad formulations have different abilities to create this transfer layer. Extensive testing has shown that the stability of the transfer layer is not only determined by the pad material but is also strongly dependent on the structure of the disc surface.

In the Volvo program, tests with the same pad formulation showed that some of the discs could withstand temperatures above 450°C without any problems while other discs, also with a fairly good particle distribution, suffered from surface scoring at temperatures below 400°C. Investigations of the disc surfaces suggested that the fraction of particles at the surface was insufficient. Rather than increasing the volume fraction of SiC in the material, which would significantly increase the material cost, Volvo used an etching process to remove aluminium from the surface, thus exposing a higher concentration of SiC. Higher material costs also excluded the use of ceramic coatings to improve the thermal stability of the disc surface. The effect on the SiC surface concentrations (in black) can be seen for the unetched and etched conditions in Figure 3-6 and Figure 3-7, respectively.

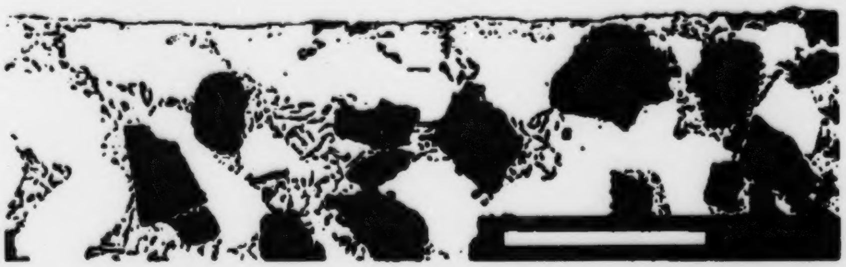

Figure 3-6 Typical Unetched MMC Disc Cross Section

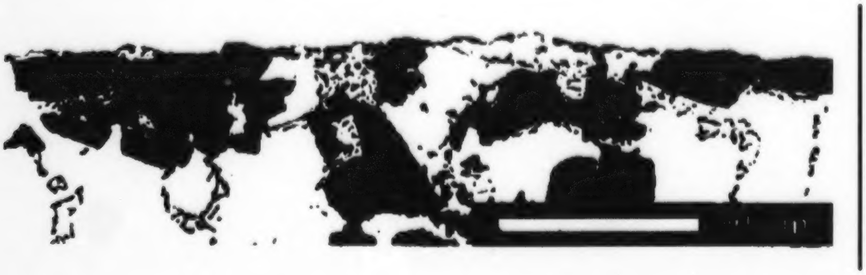

Figure 3-7 Surface Etched MMC Cross Section

An important requirement for an MMC brake is the ability to create a stable transfer layer without any special bedding-in procedures.

Volvo found that the non surface-treated discs did not fulfil this requirement. Tests showed that these discs should not be exposed to any heavy braking before the transfer layer has been developed, which could take 20 to 50 moderate brake applications, depending on the lining formulation.

For the treated discs there was no need for a special bedding-in procedure. The number of brake applications needed for the development of the transfer layer on the treated disc was typically one or two. This was seen as the effect of the treatment in increasing the surface roughness. The topography is quickly smoothed during the first brake applications when the transfer layer is being developed, shown in Figure 3-8.

Figure 3-8 Etched Disc Cross Section After First Brake Application

The increased roughness of the disc surface provided a fast buildup of a robust and stable transfer layer. The rough interface between the disc surface and the transfer layer increases the adhesion of the transfer layer and thus contributes to the increased temperature stability.

Volvo found that a maximum disc temperature of 450°C could be permitted during severe acceptance testing. The maximum surface temperature was considerably higher, but caused no particular damage on surface-treated discs.

Because of the high thermal conductivity of the MMC material, the influence of the wheel on the disc temperature was found to be considerable. Tests in the Arizona desert with the Volvo S40 showed that the front disc temperature decreased about 100°C in a stop and go test with aluminum wheels as compared to steel wheels.

The Volvo developments provide further indication that MMC can be made to physically survive the high thermal stress of vehicle applications. As the specific energy input is higher in the automotive disc application than in the truck drum, it would be anticipated that the peak temperatures would be lower in a truck application. No modelling of the Volvo design was undertaken, but it is judged that the addition of the surface etching will reduce the peak surface temperatures predicted by the CANMET model.

3.8 Conclusions

In comparison with cast iron, the SiC/Al, C/SiC and Al_2O_3/Al composite materials offer superior thermal, mechanical and physical properties, in addition to lighter weight. However, not all composites are suitable candidates for brake rotor materials for heavy vehicles. There are many properties to be considered in selecting a good composite material for brake application. For example, an important parameter to be considered is the MOT of the material. Using this parameter as a gauge, SiC/Al only offers a MOT of 450°C, whereas cast iron and C/SiC provide MOTs of 700°C and 1350°C respectively.

Results of the computer modelling work at CANMET provide temperature profiles of both disc and drum brake rotors (made with cast iron, SiC/Al and C/SiC) of a truck on a downhill descent followed by an emergency stop. These results are summarized in Table 2-6.

Table 3-6 Estimated Maximum Temperatures

Brake Type and Test	Cast Iron	SiC/Al	C/SiC
Disc Brake Rotor			
Maximum Temperature (°C) During Slow Braking	310	262	349
Maximum Temperature (°C) During Emergency Braking	593	615	713
Drum Brake Rotor			
Maximum Temperature (°C) During Emergency Braking	440	447	551

In addition, the distinction of temperature rise as a result of slow braking and emergency braking is made. When a disc brake is used and slow braking is applied, SiC/Al provides the lowest temperature of 262°C. This is understandable from the thermal conductivity standpoint. However, when emergency braking is applied, there is insufficient time for a SiC/Al brake to dissipate the heat acquired during braking, and a brake temperature of 615°C is the result.

It should be pointed out that the CANMET models were built with extremely conservative boundary conditions. That is, an assumption that almost no heat is transferred from the brake rotors to the connecting axle was made. Based on this conservative estimate and using only MOT as a criterion, it is concluded that:

- SiC/Al brake rotors (with a MOT of 450°C) might not be suitable for use in heavy vehicles. However, as mentioned earlier, Volvo work on surface treatment provides a higher MOT at the surface that the bulk MMC material and this may extend the temperature tolerance.
- CANMET-modified Al_2O_3/Al composite promised to have a higher MOT of around 600°C, but full dynamometer test data are not yet available. In a preliminary study, the CANMET-modified Al_2O_3/Al composite showed much better high temperature compressive strength than the conventional SiC/Al. Further testing is needed to evaluate the potential of such material.
- C/SiC shows the best performance with a MOT of 1350°C, and is undoubtedly technically suitable for truck applications.

A drum brake model was also used in the situation of a truck on a downhill descent followed by an emergency stop. The estimated temperatures of the drum brake if made with cast iron, SiC/Al or C/SiC are 440°C, 447°C, and 1350°C, respectively. Again, if MOT is used as a criterion, all three materials are suitable for drum brake application. The safety margin on the SiC/Al is low. However, it is believed that it will perform satisfactorily because:

- the thermal model is conservative in that it is believed to overestimate the temperatures; and
- the MOT is based on the current commercial product material's MOT and, as mentioned above, with the Volvo surface treatment technique and/or changes in the solidus material, higher effective MOTs are possible.

The modelling results on stress analysis also indicate that all three composites can sustain the stresses developed in both the disc and drum rotors during severe braking conditions. However, more careful examination of the composite properties at elevated temperatures (during the extreme emergency braking condition) should be performed to determine the suitability of such materials.

4. TECHNICAL COST ANALYSIS

4.1 Background and Approach

With dramatic advances in materials engineering and manufacturing technology, upper management decisions within the manufacturing and commodities industries have become increasingly difficult. When faced with, or indeed, developing a new material option for a component or a new method of manufacture, a decision support tool that describes the economics of adoption relative to other alternatives can be extremely useful. Besides economics, there are questions of environmental impact, customer utility and competitor activities to consider.

Technical cost modelling (TCM) is a proven decision support tool that can be of direct and timely use to the automotive and general transportation sector. TCM is a method of cost estimation that goes beyond the rules of thumb used in many manufacturing situations, which generally rely upon some factor to be multiplied by the raw material cost. In contrast to conventional cost estimation methods, TCM is more concerned with the details of the technology (thus the name), is more explicit than the methods of managerial (or "cost") accounting, and is more readily applicable than microeconomics, while incorporating principles of each.

TCM is a very effective tool to help decide whether to pursue a particular choice of material or manufacturing technology. Very explicit cost information can be generated, with sensitivity analyses, for comparison and evaluation.

In this study the manufacturing costs for brake components of class 7/8 vehicles were evaluated for drums and rotors of several materials and manufacturing processes. The particular cases evaluated are shown in Table 4-1.

For consistency, certain global assumptions were made that would apply to each process/material choice:

- A production volume of 200,000 parts per year was chosen as a reasonable market share that would reflect the potential gap between current cost factors in cast iron parts and the alternatives.
- Certain input factors, such as labour rates, depreciation times for equipment and buildings, and energy rates, were held constant throughout the analysis.
- One rotor design and one drum design were applied to all materials and processes.
- No accommodation was made for change in material properties or potential design changes that might be necessary or made available.
- The energy required for melting the aluminum and MMC, the holding times required, and the ladle travel times were all used to calculate the energy required in the production of parts in a separate model.

Table 4-1 Materials/Manufacturing Methods and Weights of Components

Material/Manufacturing System	Component Weight	
	kg	lb.
Cast iron drum	50.8	112
Cast iron rotor	27.2	60
NRCan drum – with preform infiltration	21.3	47
NRCan rotor – with preform infiltration	11.8	26
MMC drum – from MMC billet	21.3	47
MMC rotor – from MMC billet	11.8	26
MMC drum – with MMC mixed on site	21.3	47
MMC rotor – with MMC mixed on site	11.8	26
Carbon/Silicon Carbide ceramic rotor	10	22

For each material/technology evaluated, a cost model was prepared using input variables such as the global ones listed above, as well as specific material costs, recoveries and rejection rates, equipment costs and capacities, labour requirements and energy consumptions, among others. Generally, these models follow the process route in the manufacture of the component. The process flow diagram, cost summary, and sensitivity analysis for each material/technology are presented in this chapter. It should be noted that all costs are presented in US dollars.

The material costs were calculated through part size, recoveries and rejection rates within each process step. By determining the number of direct operators for each process/machine and indirect labour for the manufacturing facility, a total labour cost was calculated and distributed over the production volume. Likewise the capital investment, energy costs, and other costs involved in the manufacture of the component were calculated, as was the sensitivity to changes in input variables.

4.2 Analysis and Results

4.2.1 Cast Iron Drum and Rotor Models

The modelling of the cast iron components started from the delivery of the rough casting from the foundry. Starting with pig iron as a raw material and following through the casting process was deemed to be inappropriate for the following reasons:

- The facilities that produce castings for the automotive industry, and the heavy truck market in particular, have very large capacities.
- These facilities are not generally integrated with the finishing of the product.
- We received what is believed to be an accurate cost figure for the drum casting as received by the manufacturer (US$28), and an accurate price quote for rotor castings (US$0.40/lb.) from a foundry that specializes in braking components

Most of the cost information was gleaned from conversation with manufacturers of drums for OEMs and the aftermarket.

Cast Iron Drum

The process flow for manufacturing cast iron drums is shown in Figure 4-1. The cost of manufacturing the drum from rough castings purchased from an iron foundry is estimated to be US$32.21. A summary of the fixed and variable costs is given in Table 4-2. The sensitivity analysis summarized in Figure 4-2 displays the importance of the raw material cost, which is to be expected in this case. By purchasing the rough castings, and considering them as raw materials, the value added is quite high at this point in the supply chain. We can then presume that the raw materials cost would represent a large portion of the total cost of manufacture, and that the total cost would show an acute sensitivity to change in that raw materials cost.

Table 4-2 Cost Summary for Cast Iron Drum

CAST IRON BRAKE DRUM - CLASS 7/8 TRUCK

		Cost per part (US$/part)	Cost per year (US$/year)	Investment
Variable Costs				
	Materials	$28.43	$5,685,047	
	Direct Labour	$2.18	$435,456	
	Energy	$0.10	$20,000	
	Total Variable Cost	$30.70	$6,140,503	
Fixed Costs				
	Equipment	$0.47	$94,000	$1.410,000
	Tooling	$0.35	$69,560	
	Building	$0.15	$30,000	
	Indirect Labour	$0.54	$108,000	
	Total Fixed Cost	$1.51	$301,560	
	GRAND TOTAL	**$32.21**	**$6,442,063**	

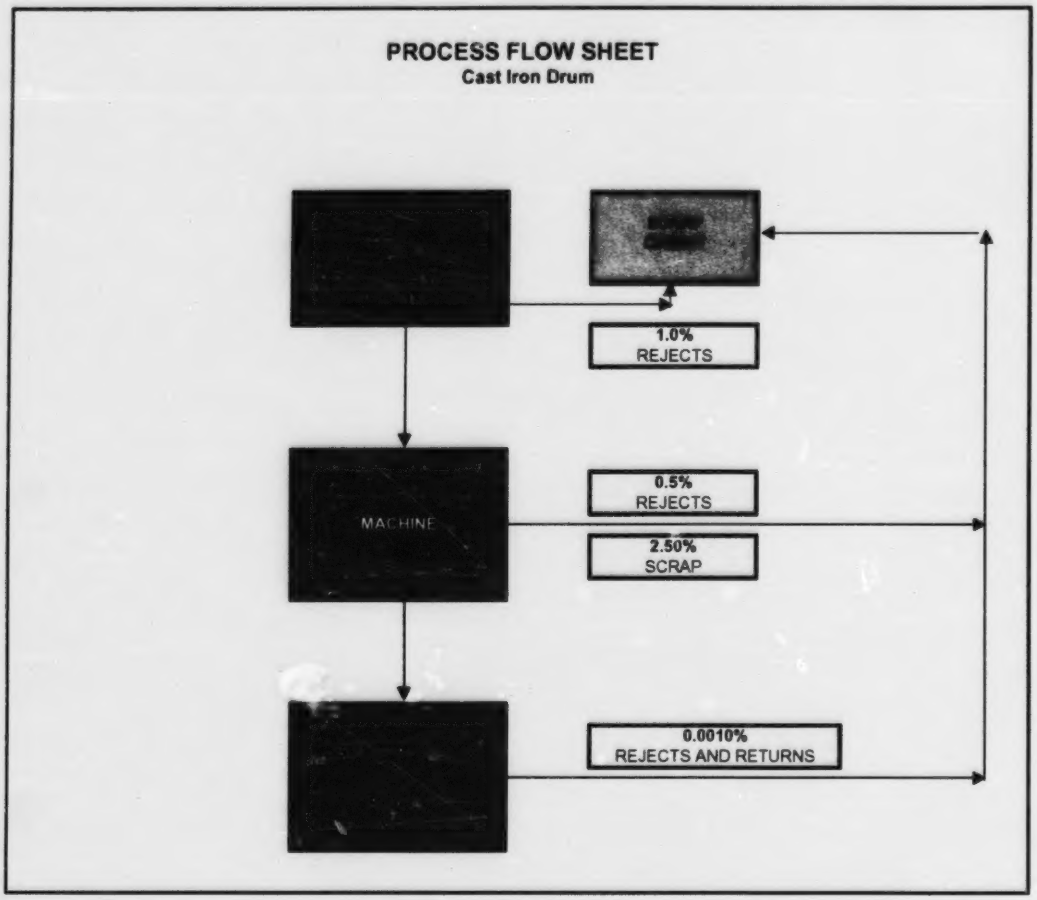

Figure 4-1 Process Flow Diagram for Cast Iron Drum Manufacture

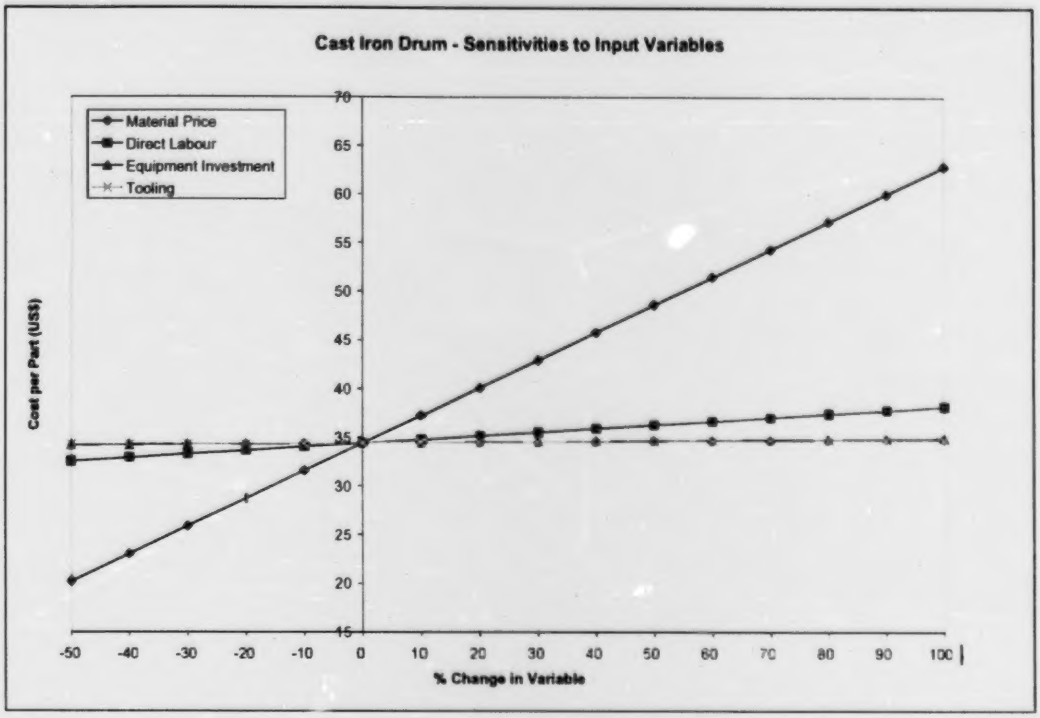

Figure 4-2 Sensitivity Chart for Cast Iron Drum

Cast Iron Rotor

The process flow for manufacturing cast iron rotors is shown in Figure 3.3 and the summary of the costs model is given in Table 4-3. The cost of the rough casting in this case is estimated to be US$0.40/lb. This estimate was provided by Waupaca Foundry, based upon our 200,000 unit production level. The total estimated manufacturing cost for the cast iron rotor is US$31.82. The manufacturing process is the same as for the drum, so the sensitivities shown in Figure 3.4 and the relative cost breakdowns are very similar.

Table 4-3 Cost Summary for Cast Iron Rotor

CAST IRON BRAKE ROTOR - CLASS 7/8 TRUCK

		Cost per part (US$/part)	**Cost per year** (US$/year)	**Investment**
Variable Costs				
	Materials	$28.72	$5,743,058	
	Direct Labour	$1.91	$381,024	
	Energy	$0.10	$20,000	
	Total Variable Cost	$30.72	$6,144,082	
Fixed Costs				
	Equipment	$0.14	$27,333	$410,000
	Tooling	$0.27	$54,560	
	Building	$0.15	$30,000	
	Indirect Labour	$0.54	$108,000	
	Total Fixed Cost	$1.10	$219,893	
	GRAND TOTAL	**$31.82**	**$6,363,975**	

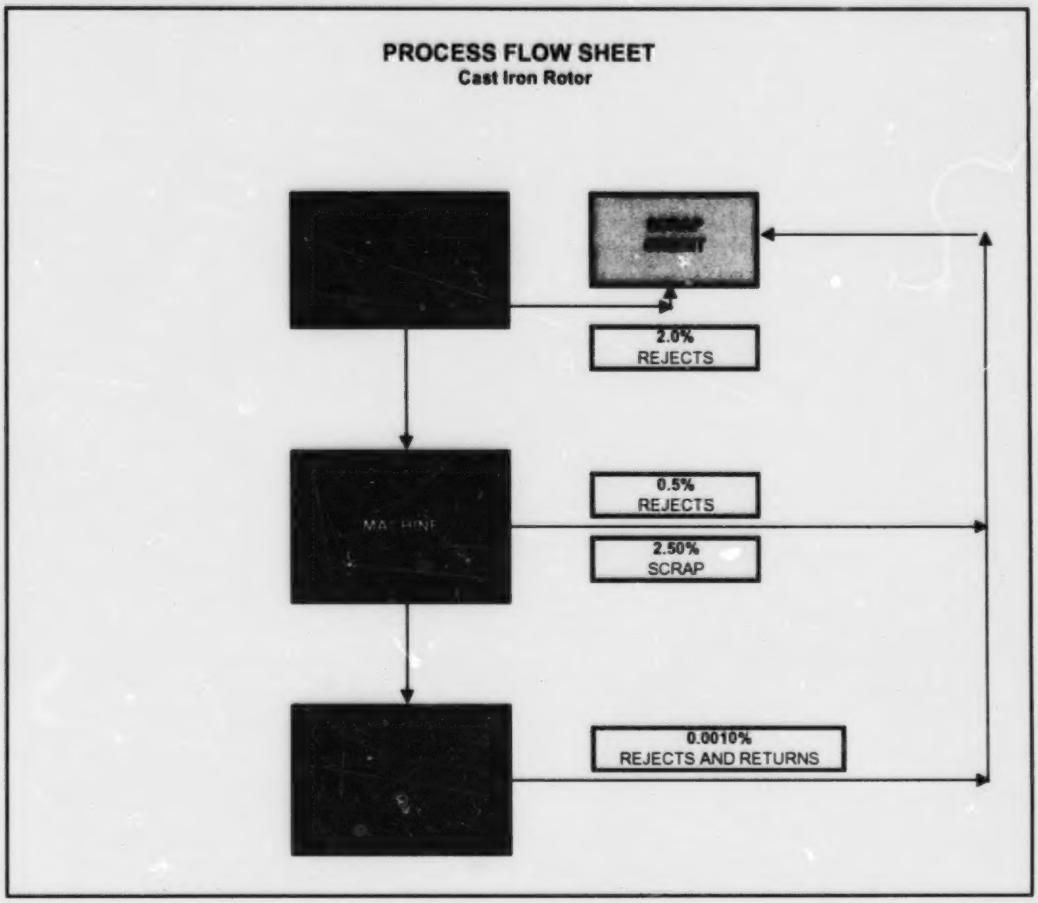

Figure 4-3 Process Flow Diagram for Cast Iron Rotor Manufacture

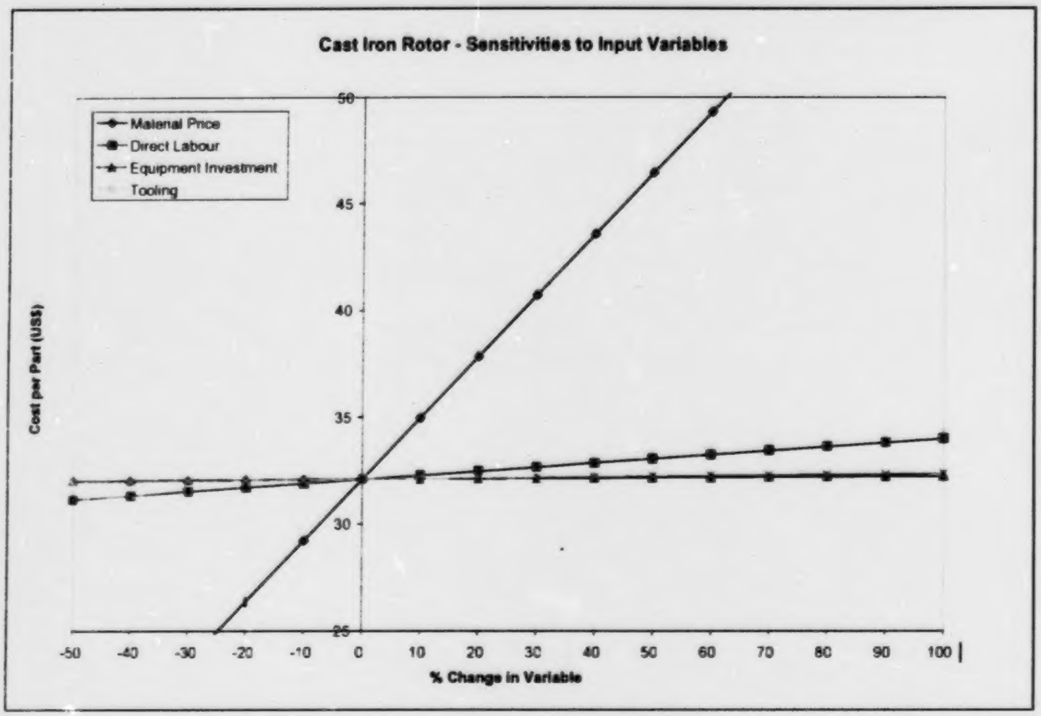

Figure 4-4 Sensitivity Chart for Cast Iron Rotor

4.2.2 NRCan Proprietary Method with Preform Infiltration

Researchers at NRCan's CANMET have developed a method for selective positioning of the MMC material in a component by placing a fibre preform where the MMC is most desirable (in the contact areas) and having molten aluminum squeeze cast into the part die, resulting in the infiltration of the preform. Using aluminum with an infiltrated preform gives a (relatively) low raw material cost for the bulk of the section, with the required strength and wear resistance in the contact areas.

CANMET provided much of the information used in these models, including the process parameters and the preform cost estimates. Because of the proprietary nature of the process and the lack of widespread application, some of the data in the model were estimated and gleaned from other models and other sources of information.

The manufacturing flow diagram for the NRCan process for drums and rotors is shown in Figure 4-5.

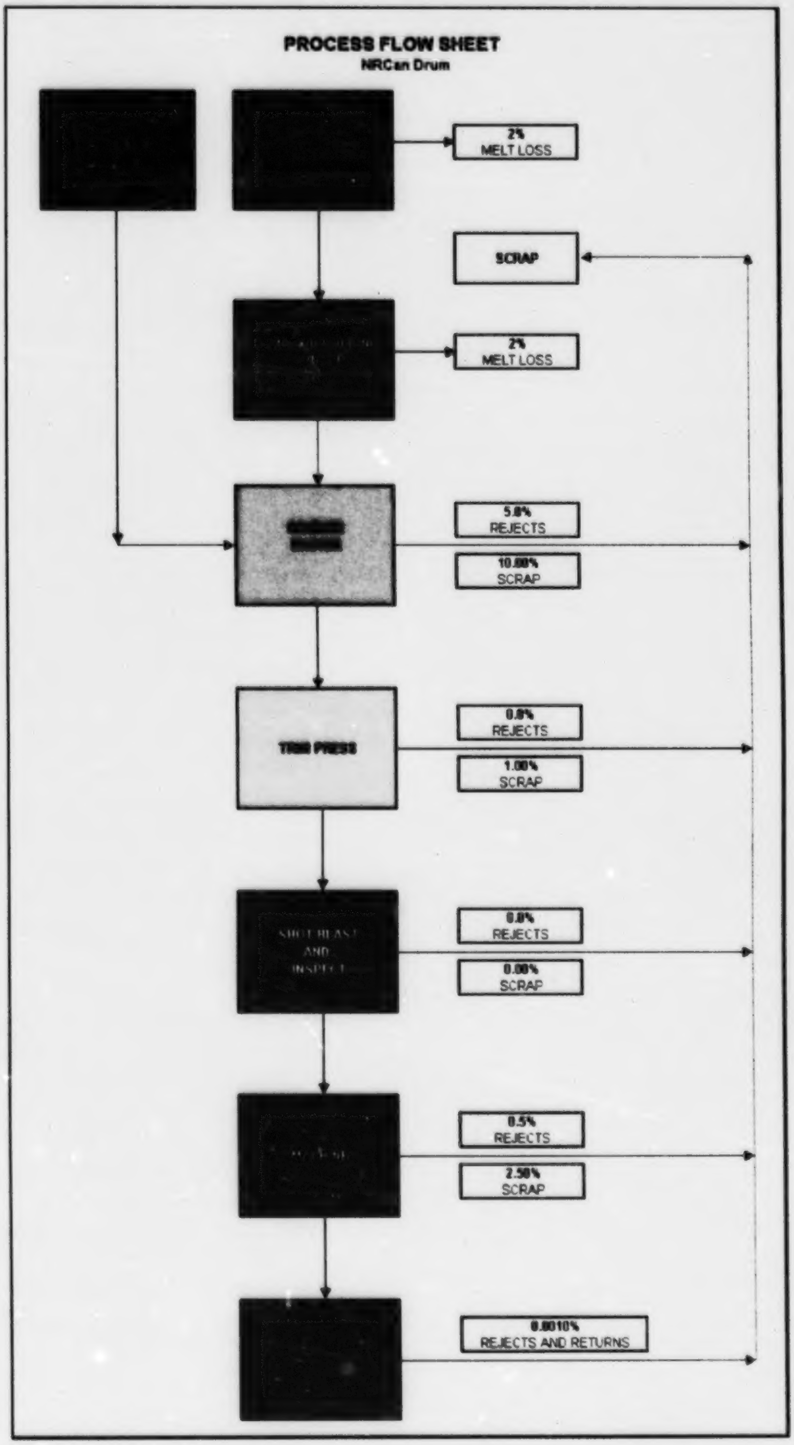

Figure 4-5 Process Flow Diagram for NRCan Manufacture of Drums and Rotors

NRCan Drum

In the case of the drum, where a large proportion of the part is aluminum alloy, some material cost reduction can be achieved and machining costs can also be reduced, since much of the finishing (hole drilling, mating surface preparation) can be performed on the aluminum alloy and not on the harder MMC material, which requires different, more expensive tooling and longer machining times. There remains the concern that the structure may need to be reinforced to meet strength requirements, increasing the amount of material required and, therefore, the manufacturing cost.

Table 4-4 and Figure 4-6 show the cost summary and the results of the sensitivity analysis, respectively, for the drum manufacture. The total manufacturing cost estimate for the drum is US$56.86, which is US$24.65 higher than the cost of a cast iron drum. The cost is once again most sensitive to change in raw material cost, both in the form of aluminum and preform. The preform cost estimate was based on CANMET's experience purchasing preforms for smaller rotors and other preform geometries.

Table 4-4 Cost Summary for NRCan Drum

NRCan BRAKE DRUM - CLASS 7/8 TRUCK

		Cost per part (US$/part)	**Cost per year** (US$/year)	**Investment**
Variable Costs				
	Materials	$43.21	$8,642,495	
	Direct Labour	$6.48	$1,296,000	
	Energy	$3.75	$750,000	
	Total Variable Cost	$53.44	$10,688,495	
Fixed Costs				
	Equipment	$0.75	$149,889	$2,245,000
	Tooling	$0.59	$118,824	
	Building	$0.50	$100,000	
	Indirect Labour	$1.58	$315,000	
	Total Fixed Cost	$3.42	$683,713	
	GRAND TOTAL	**$56.86**	**$11,372,208**	

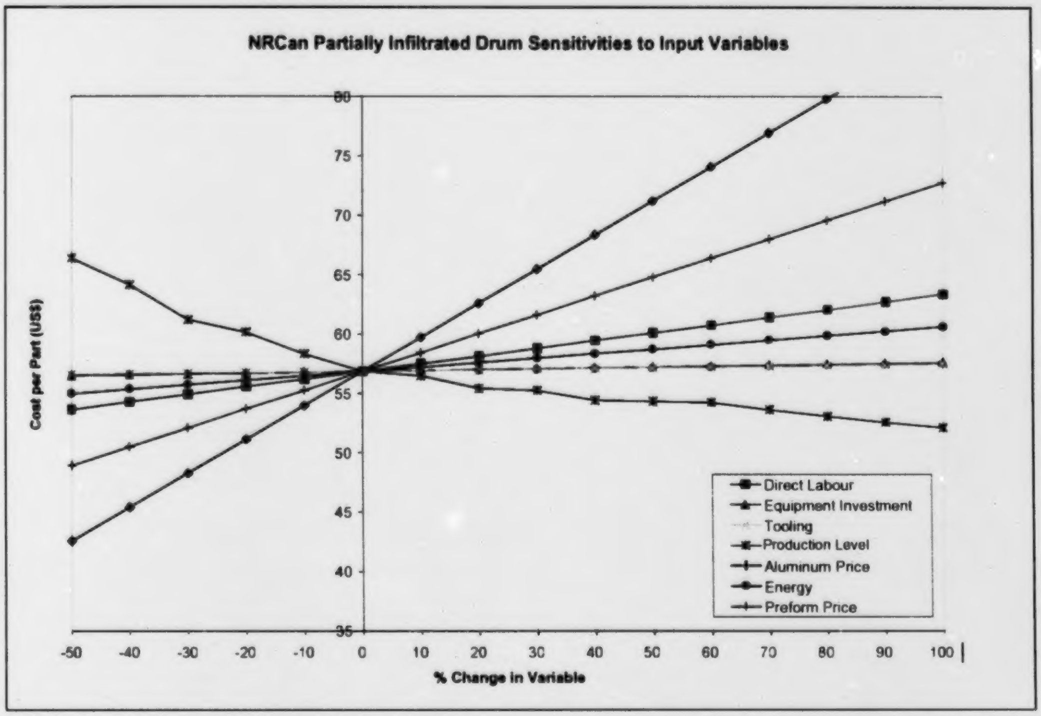

Figure 4-6 Sensitivity Chart for NRCan Drum

NRCan Rotor

The rotor may be less susceptible than the drum to design problems because aluminum alloy is used as the bulk material, but since a higher proportion of the part is occupied by the more expensive preform, the part is relatively more costly.

Table 4-5 and Figure 4-7 show the cost summary and the sensitivity chart for the NRCan rotor.

Table 4-5 Cost Summary for NRCan Rotor

NRCan BRAKE ROTOR - CLASS 7/8 TRUCK

	Cost per part (US$/part)	Cost per year (US$/year)	Investment
Variable Costs			
Materials	$27.52	$5,504,137	
Direct Labour	$6.48	$1,296,000	
Energy	$3.75	$750,000	
Total Variable Cost	$37.75	$7,550,137	
Fixed Costs			
Equipment	$0.42	$83,222	$1,245,000
Tooling	$0.52	$103,824	
Building	$0.50	$100,000	
Indirect Labour	$1.58	$315,000	
Total Fixed Cost	$3.01	$602,046	
GRAND TOTAL	**$40.76**	**$8,152,183**	

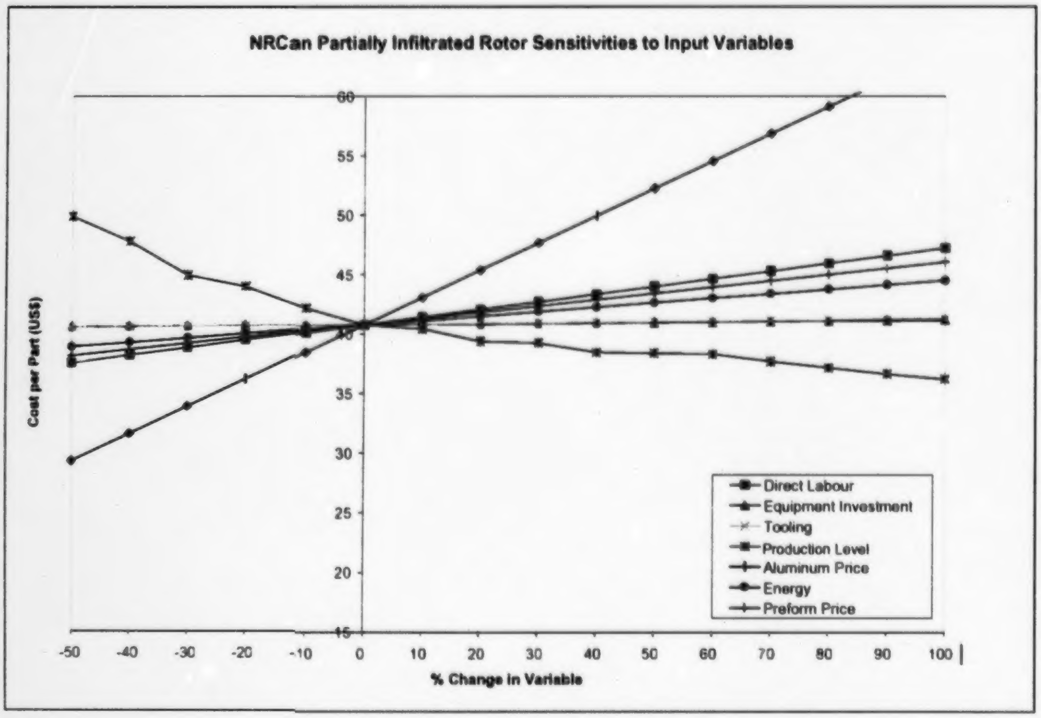

Figure 4-7 Sensitivity Chart for NRCan Rotor

4.2.3 Aluminum/Silicon Carbide MMC from Billet

The production of homogeneous MMC components in this case consists of purchasing MMC material in the form of billet (the price was estimated, based on the production level, by Alcan Engineered Cast Products), remelting, and casting the part by one of several available methods.

MMC Drum from Billet

It was the opinion of the people interviewed during the study[12] that the best (i.e., most economical) way to cast the drums in production quantity would be to use a permanent mould system. Luc Paris of Paber supplied an estimate of capital required and expected capacity, and provided a history of the trial castings that were produced by Paber for one of the large players in the truck brake component market.

It is assumed in this model that the scrap can be recycled in the plant with the use of a reprocessor. If this were not the case, the per-part cost would increase substantially. As modelled, the cost of the drum is estimated at US$77.79, which is more than double that of the cast iron drum. The cost is extremely sensitive to the material cost once again, as would be expected. The cost summary, sensitivity chart and process flow diagram for the MMC drum from billet follow as Table 4-6, Figure 4-8 and Figure 4-9, respectively.

Table 4-6 Cost Summary for PM MMC Drum from Billet

DURALCAN MMC BRAKE DRUM - CLASS 7/8 TRUCK

Permanent Mould, MMC from Billet

		Cost per part (US$/part)	**Cost per year** (US$/year)	**Investment**
Variable Costs				
	Materials	$67.94	$13,588,203	
	Direct Labour	$3.81	$762,048	
	Energy	$2.28	$455,768	
	Total Variable Cost	$74.03	$14,806,019	
Fixed Costs				
	Equipment	$1.32	$264,333	$3,965,000
	Tooling	$0.24	$48,040	
	Building	$1.25	$250,000	
	Indirect Labour	$0.95	$189,000	
	Total Fixed Cost	$3.76	$751,373	
	GRAND TOTAL	**$77.79**	**$15,557,392**	

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[12] David Weiss of Eck Foundry, and Luc Paris of Paber Aluminum.

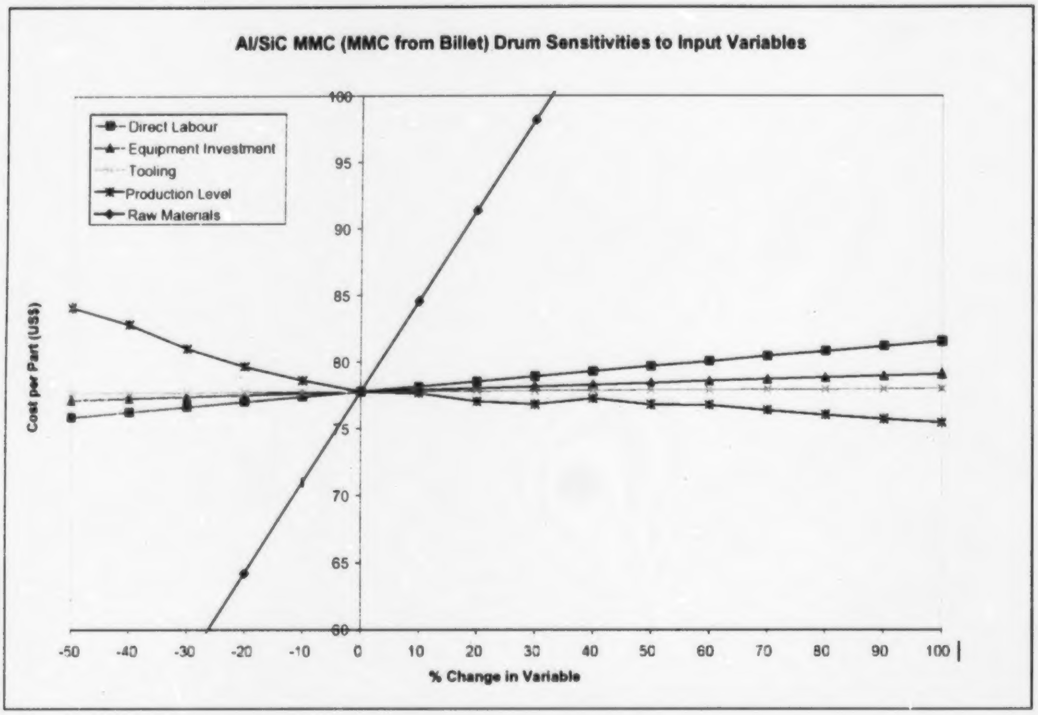

Figure 4-8 Sensitivity Chart for PM MMC Drum from Billet

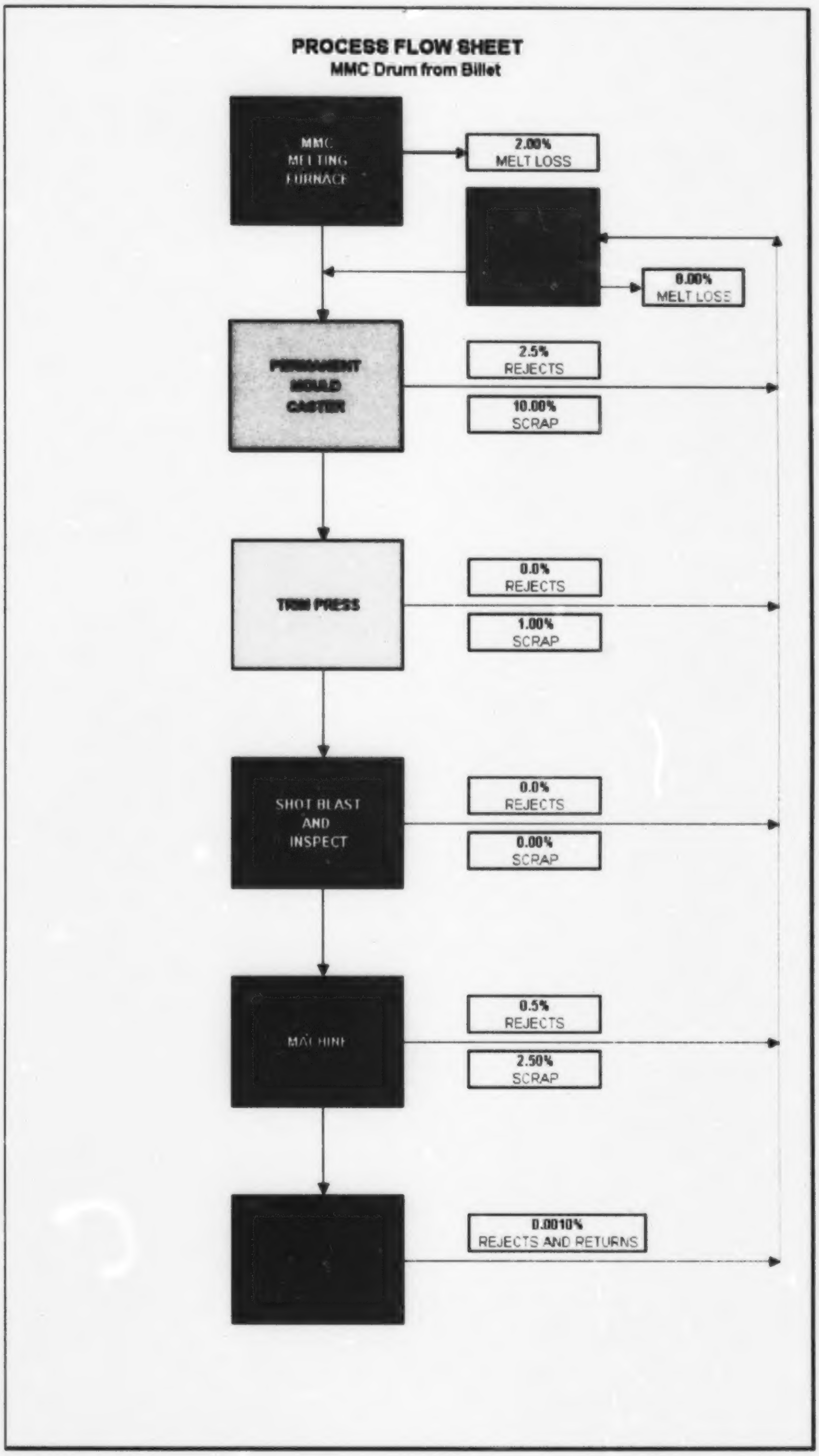

Figure 4-9 Process Flow Diagram for PM MMC Drum from Billet

MMC Rotor from Billet

For this smaller component we assumed (from discussions and earlier work) that it would be possible/feasible to use high-pressure die casting (HPDC). We again assumed that the scrap could be recycled. The process flow chart for using HPDC to produce the component is very similar to the permanent mould process, since the only change is in the casting process itself. The cost summary, sensitivity chart and process flow diagram follow as Table 4-7, Figure 4-10 and Figure 4-11, respectively. The cost of manufacture for the MMC rotor produced from billet is estimated at US$46.66.

Table 4-7 Cost Summary for HPDC MMC Rotor from Billet

Al/SiC MMC BRAKE ROTOR - CLASS 7/8 TRUCK
HPDC, MMC from Billet

		Cost per part (US$/part)	Cost per year (US$/year)	Investment
Variable Costs				
	Materials	$38.06	$7,612,254	
	Direct Labour	$3.54	$707,616	
	Energy	$1.52	$303,234	
	Total Variable Cost	$43.12	$8,623,103	
Fixed Costs				
	Equipment	$0.61	$121,000	$1,815,000
	Tooling	$0.75	$149,707	
	Building	$1.25	$250,000	
	Indirect Labour	$0.95	$189,000	
	Total Fixed Cost	$3.55	$709,707	
	GRAND TOTAL	**$46.66**	**$9,332,810**	

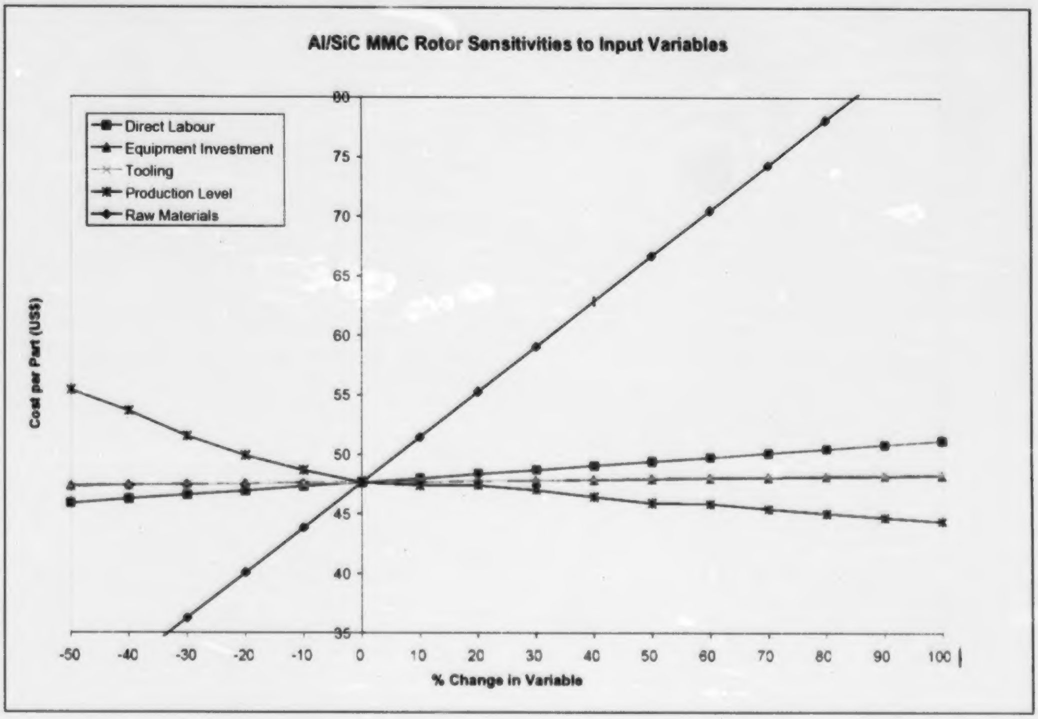

Figure 4-10 Sensitivity Chart for HPDC MMC Rotor from Billet

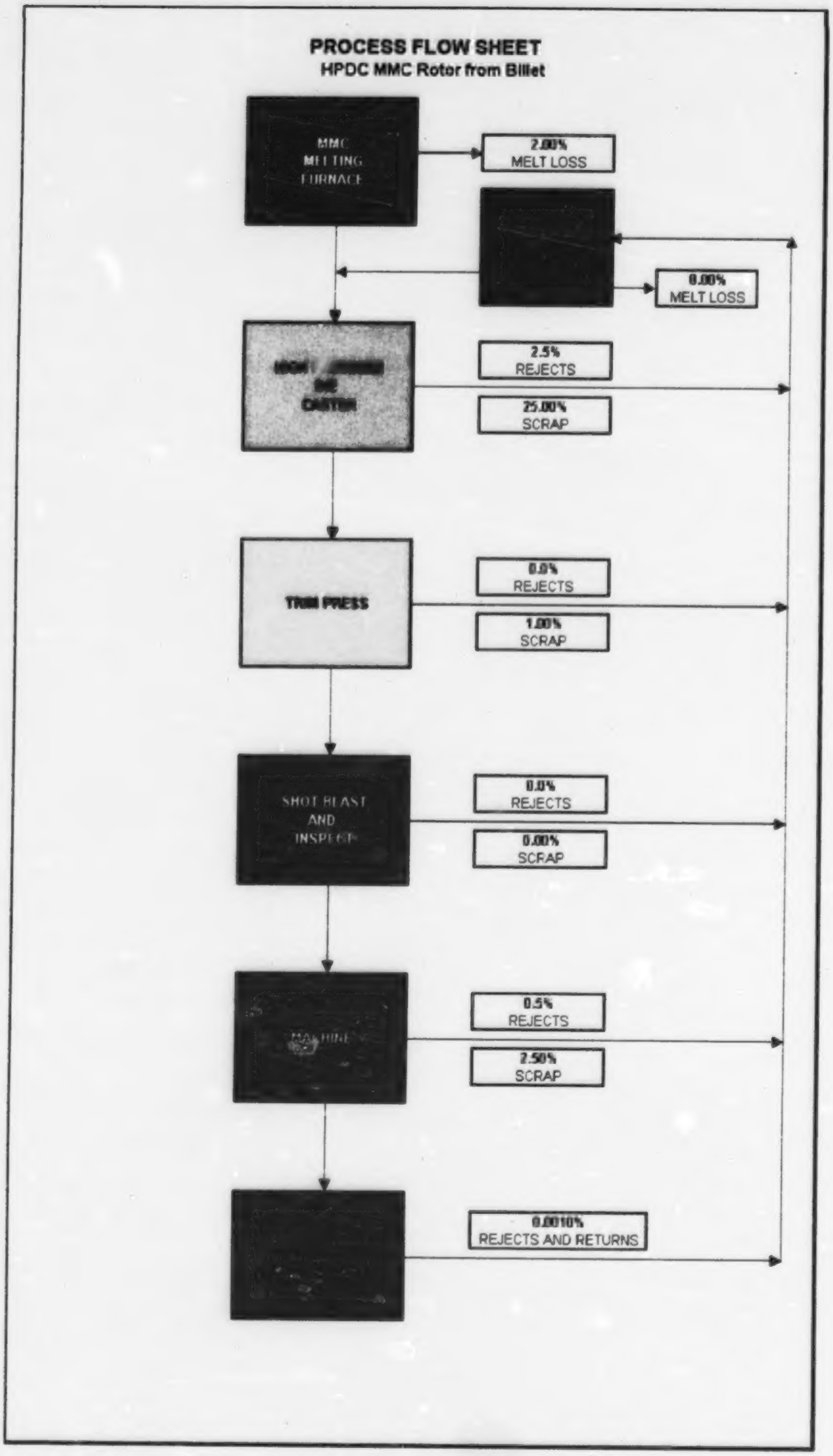

Figure 4-11 Process Flow Diagram for HPDC MMC Rotor from Billet

4.2.4 Aluminum/Silicon Carbide MMC with Integrated MMC Mixing

The idea of mixing the MMC on site, thus eliminating the casting/remelting steps (and consequent costs) in the supply chain has garnered considerable interest in the marketplace, with both Duralcan and MC-21 (MMC suppliers) exploring opportunities to set up production systems using proprietary processes to ensure quality product out of "mini" melting/mixing units. As is the case with the NRCan process, these are also proprietary processes, making the estimation of production costs more uncertain.

MMC Drum with Integrated MMC Mixing

The drum was modelled once again using permanent mould casting, this time using MMC mixed on site, and with scrap recycled in a scrap reprocessor. When compared with the process using billet, the total manufacturing cost dropped with the elimination of intermediary steps to US$61.43. Still, the raw material cost is the key factor in the sensitivity of total manufacturing cost. The cost summary, sensitivity chart and process diagram for the MMC drum produced with integrated MMC mixing follow in Table 4-8, Figure 4-12 and Figure 4-13.

Table 4-8 Cost Summary for MMC Drum with Integrated MMC Mixing

Al/SiC MMC BRAKE DRUM - CLASS 7/8 TRUCK
Permanent Mold, MMC mixed on site

		Cost per part (US$/part)	**Cost per year** (US$/year)	**Investment**
Variable Costs				
	Materials	$46.40	$9,280,734	
	Direct Labour	$3.54	$707,616	
	Energy	$2.69	$537,185	
	Total Variable Cost	$52.63	$10,525,535	
Fixed Costs				
	Equipment	$1.57	$314,333	$4,715,000
	Tooling	$0.17	$33,040	
	Building	$1.25	$250,000	
	Indirect Labour	$1.35	$270,000	
	Total Fixed Cost	$4.34	$867,373	
	GRAND TOTAL	**$56.96**	**$11,392,908**	

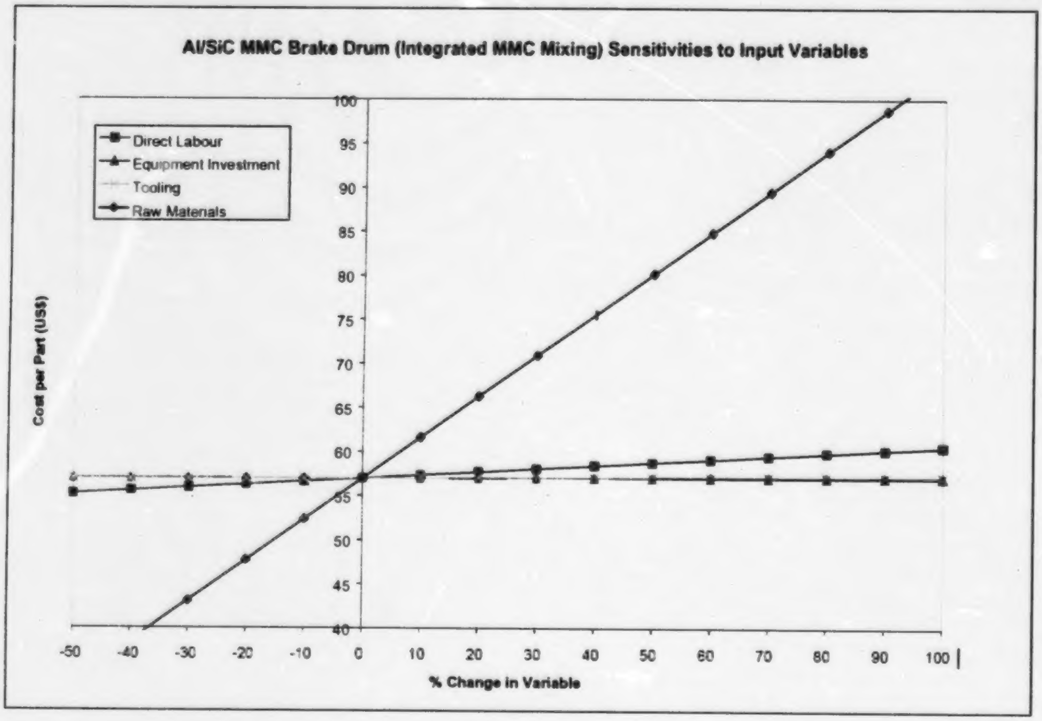

Figure 4-12 Sensitivity Chart for MMC Drum with Integrated MMC Mixing

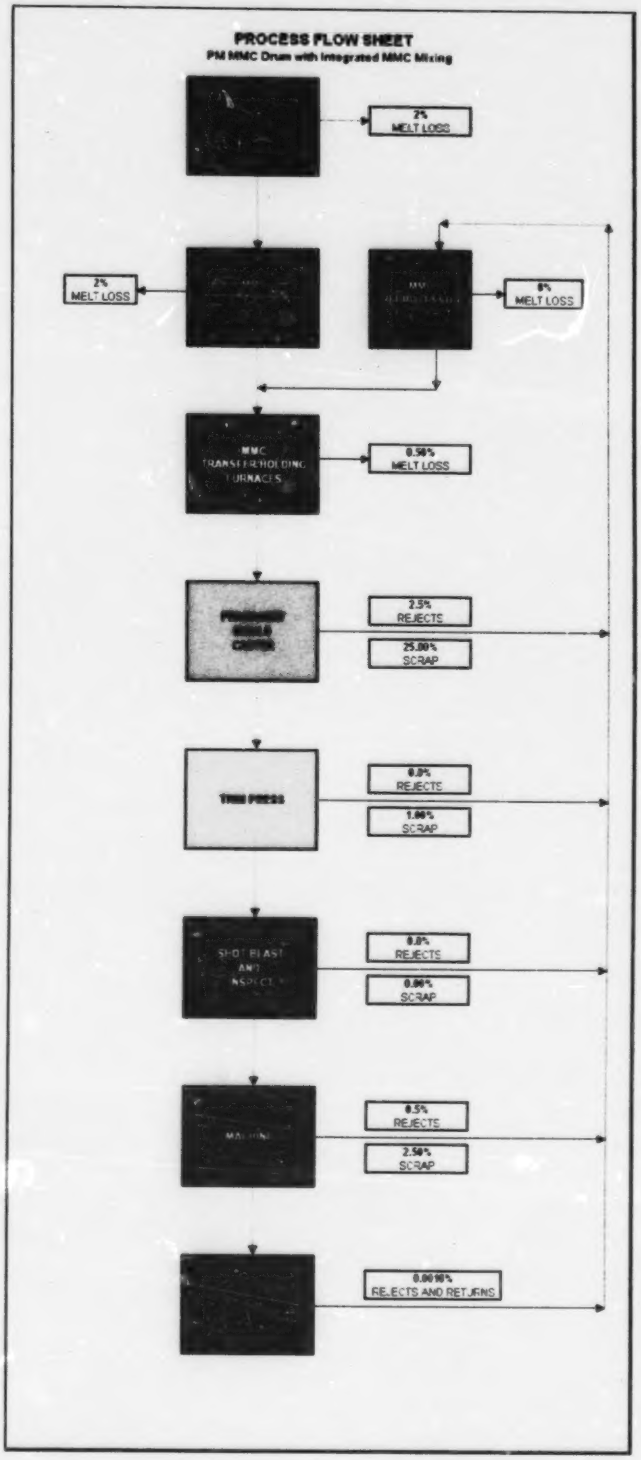

Figure 4-13 Process Flow Diagram for MMC Drum with Integrated MMC Mixing

MMC Rotor with Integrated MMC Mixing

Mixing MMC in a small unit next to the HPDC machine is a process that is being investigated on several fronts, and has potential to reduce the manufacturing cost of a part considerably when compared with a billet-based process. In the case of the rotor, the part cost was estimated at US$35.82, which represents a US$4.00 premium over the cast iron rotor. The cost summary, process diagram, and sensitivity chart for manufacturing the MMC rotor with integrated MMC mixing follow in Table 4-9, in Figure 4-14, and Figure 4-15.

Table 4-9 Cost Summary for MMC Rotor with Integrated MMC Mixing

Al/SiC MMC BRAKE ROTOR - CLASS 7/8 TRUCK
HPDC, MMC mixed on site

		Cost per part (US$/part)	Cost per year (US$/year)	Investment
Variable Costs				
	Materials	$26.22	$5,244,117	
	Direct Labour	$3.54	$707,616	
	Energy	$1.52	$304,758	
	Total Variable Cost	$31.28	$6,256,490	
Fixed Costs				
	Equipment	$1.19	$237,667	$3,565,000
	Tooling	$0.75	$149,707	
	Building	$1.25	$250,000	
	Indirect Labour	$1.35	$270,000	
	Total Fixed Cost	$4.54	$907,373	
	GRAND TOTAL	**$35.82**	**$7,163,864**	

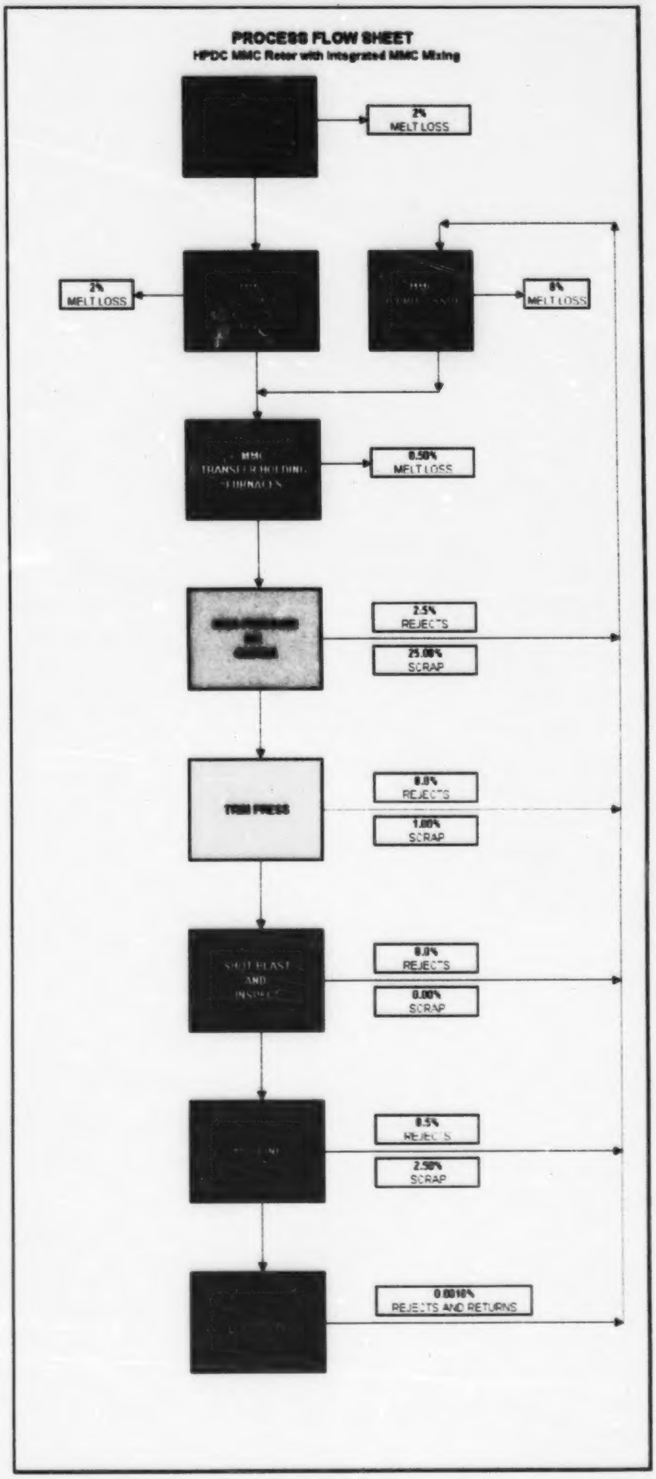

Figure 4-14 Process Flow Diagram for MMC Rotor with Integrated MMC Mixing

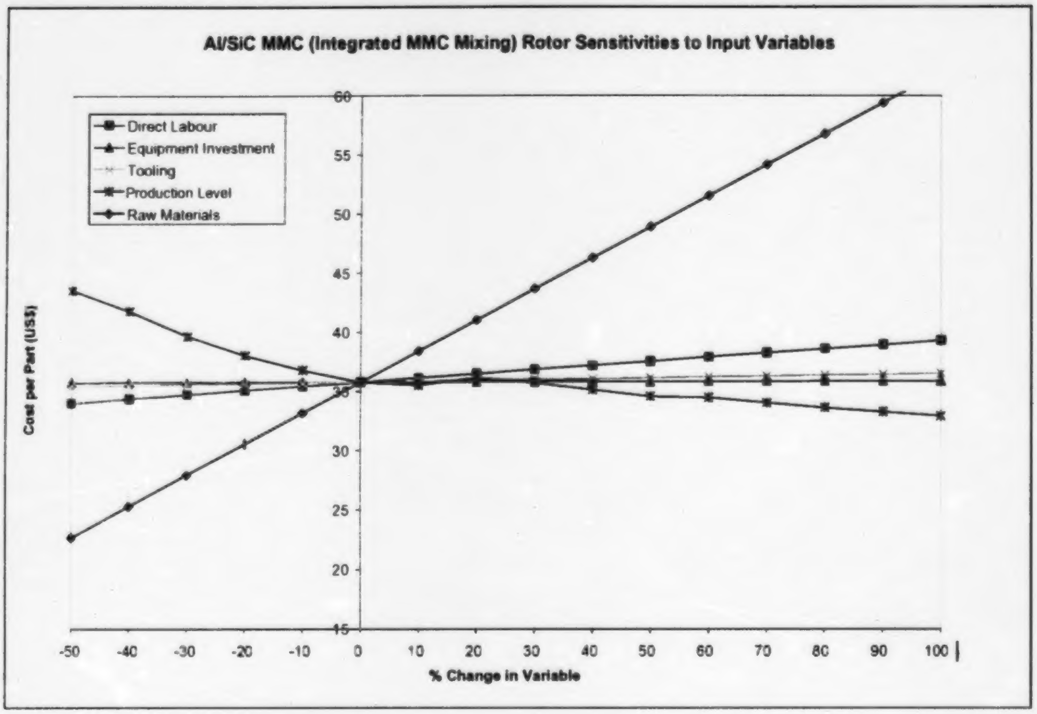

Figure 4-15 Sensitivity Chart for MMC Rotor with Integrated MMC Mixing

4.2.5 Ceramic (C/SiC) Rotors

The modelling of the manufacture of a ceramic brake rotor proved to be a considerable challenge because of the relative novelty of the technology in the automotive industry (Porsche now offers C/SiC composite brake rotors as a very expensive option (over US$11,000) on the 911 Turbo, and as standard equipment on the 911 GT-2). The production level is still small enough that the components are produced in little more than a laboratory environment, making the modelling of manufacturing cost very difficult. The cost of raw materials alone may make the C/SiC rotors prohibitive for use in the heavy truck market. The price for carbon fibres, used in the preliminary stage prior to carbonization and silicon impregnation, is currently in the region of US$6 to $8, with a decrease to perhaps US$4.50 to $6 foreseen with the introduction of new products, given very large volumes.[13] This would still place the raw material cost of a brake rotor at approximately 4.5 times the complete manufacturing cost of the same rotor in cast iron (once again assuming no design changes, simply a material density substitution). Using a capital investment and labour and energy inputs similar to that for the integrated MMC manufacture, a rotor cost of US$156 was calculated.

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[13] This estimate was provided in a discussion with Buster Pipton, National Sales Manager for TORAYCA Carbon Fibers, a large supplier of carbon fibres.

5. CONCLUSIONS AND DEVELOPMENT OPTIONS

5.1 Conclusions

The potential for MMC materials to be used in truck brake drums and rotors can be assessed based on two criteria – performance and price. The performance criterion was assessed by the analysis of the physical properties and performance of the three candidate materials – MMC, the CANMET composite and C/SiC. The criteria are mutually exclusive (i.e., both criteria must be met for the product to be viable).

The conclusion of the physical analysis indicated that the aluminum-based materials would be applicable for truck drum brake applications but not for truck disc brake applications. The high temperature of the disc brakes was determined to exceed the MOT of the two aluminum materials. The C/SiC was assessed to be applicable for both configurations.

The market analysis indicated that over 90 percent of the current trucks use drum brakes but that the use of disc brakes is expanding rapidly in Europe. This trend may extend to North America but probably not without a strong regulatory push or significant price advantages. The market is very sensitive to product price, however, and will pay a premium for weight or maintenance cost reductions. The analysis estimated that a high penetration of both the new vehicle market and aftermarket could be expected for lighter weight brakes if the cost differential was below C$3/kg. The market acceptance drops precipitously after this cost range but there are segments of the market that will pay high differentials. It is estimated that approximately 20 percent of the market would be interested in products that cost more that C$10/kg.

The results of the manufacturing cost analyses for select drum and rotor designs for the various materials/technologies modelled are summarized in Table 5-1. In all cases, the cost is most sensitive to the raw material cost. The analysis shows that none of the MMC methods compared favourably on a cost-per-kilogram basis with the cost of manufacturing the components in cast iron. This is especially clear in the case of the drum, a part with a greater raw material requirement and fewer casting details.

The price penalty for the materials is significantly less if it is assessed on the basis of the cost per mass saved – a number to which the market is sensitive. These values are shown in the last two columns of Table 5.1. For the drum configurations, the lowest incremental costs are achieved (C$1.3/kg of weight saved) either with the preform or MMC mixed on site models. The premixed MMC is significantly higher at C$2.3/kg. For rotors, the costs are lowest ($0.4/kg) for the mix on site option. The preform method is estimated at slightly over twice this cost at C$0.9/kg and the premixed MMC would cost C$1.5/kg. Finally the C/SiC material has the highest cost at C$10.9/kg.

These costs are at the "factory gate" and must be inflated to obtain estimates of the retail price differentials. There are a variety of factors that affect the size of the inflator, but it is generally in the range of 2 to 2.5 for automotive parts. Thus, many of the material options in the drum configuration come close to meeting the C$3/kg target required for wide market penetration.

Table 5-1 Materials/Manufacturing Methods and Costs of Manufacture

Material/Manufacturing System	Cost of Manufacture/Unit		Unit Weight		C$/kg	Cost/ Wt Saved (C$/kg)	
	US$	C$	lb.	kg		Drum	Rotor
Cast iron drum	$32.2	$48.8	112	50.8	$1.0		
Cast iron rotor	$31.8	$48.2	60	27.2	$1.8		
NRCan drum – with preform infiltration	$56.9	$86.2	47	21.3	$4.0	$1.3	
NRCan rotor – with preform infiltration	$40.8	$61.8	26	11.8	$5.2		$0.9
MMC drum – from MMC billet	$77.8	$117.9	47	21.3	$5.5	$2.3	
MMC rotor – from MMC billet	$46.7	$70.7	26	11.8	$6.0		$1.5
MMC drum – with MMC mixed on site	$57.0	$86.3	47	21.3	$4.0	$1.3	
MMC rotor – with MMC mixed on site	$35.8	$54.3	26	11.8	$4.6		$0.4
Carbon/Silicon Carbide ceramic rotor	$156.0	$236.4	22	10.0	$23.6		$10.9

It is thus concluded that the use of aluminum MMC – either in homogeneous mixture with SiC or as a composite with Al_2O_3 – has a high probability of being both physically and financially viable for the drum brake market.

5.2 Development Requirements and Strategy

The successful further development of heavy truck brake drums and rotors requires a large number of connected factors and resources. This section will describe these requirements and lay out a possible development strategy for the commercialization of heavy-duty brakes made of the new materials.

There are three major issues that must be dealt with:

1. The production of new prototype drums and their testing, which will ensure that the results of the physical modelling are accurate and that the product will sustain the thermal and stress environments of a heavy-duty truck or bus.

2. The strategic connection with a drum manufacturer, which will ensure that the product, if it passes the technical tests, can be introduced to the market in an efficient and effective manner.

3. Confirmation of the financial models. After the technical evaluation has been successful, more detailed analysis of the production cost options must be reviewed to ensure that the product cost base as modelled is correct.

This section focusses on outlining the additional tasks that must be undertaken to validate the first issue above, physical performance, and on the choices available for OEM and supplier relationships for product development.

5.2.1 Physical Performance Testing

A series of steps must be undertaken to develop, test and qualify the feasibility of using MMC or the NRCan composite material in a drum brake. The testing must validate, first in controlled laboratory testing and then on the test track, the physical reliability and durability of the MMC-based drum product.

For the MMC material, the material, casting techniques and casting facilities are already in existence. However, for the NRCan composite material, a drum design must be fabricated and tested for its physical characteristics to confirm the estimations used in the analysis model. It must then be cast by NRCan using a custom-built squeeze cast mould. Thus, two distinct evaluation paths are described in the following text.

MMC-Based Design

The MMC-based design can be based on the material, casting and finishing techniques used previously by Hayes Lemmerz and Volvo. The most expeditious route for this option is to work in collaboration with the companies and suppliers that have been involved in the previous efforts and thus this is the focus of this path. The work steps are outlined below:

1. Develop Drum Design

Hayes Lemmerz used the Eck Foundry in Wisconsin for its MMC casting and Eck is still in retention of the casting mould that was used. Hayes Lemmerz will allow the use of its mould design. Some minor modifications to the existing pattern will be required (the interior corner radius will need to be increased to reduce the stress concentration) and can be undertaken with the assistance of Duralcan design specialists and the failure analysis documents for the previous trial of the material by Hayes Lemmerz.

2. Cast and Prepare Drum

The Eck foundry would be used for the prototype castings. Volvo Technology Centre, which developed the surface etching techniques, has conceptually agreed to provide information on its processes. The process is relatively simple and involves the use of an acid wash on the wearing surface. For the prototypes, this etching would be done manually. After casting and etching, material analysis of cross sections would be undertaken to quantify the structural changes created by the etching process.

3. Select Friction Pad Material

The previous Hayes Lemmerz test program used friction pad material developed by TMD Friction Ltd. That company has been approached for participation in this development project and is "conceptually" willing to provide friction material design and product for the evaluation program. The formulation of the correct friction material is as key to the success for the product as is the drum casting material. The previous material appeared to behave well under the laboratory testing program but the test track testing was too short (as a result of catastrophic failure of the drum) for any friction material performance conclusions.

4. Conduct Brake Dynamometer Testing

Following the fabrication and etching of the prototype drums, a performance evaluation needs to be performed under controlled laboratory conditions. This is achieved through the use of brake dynamometer testing in which the brake assembly is installed on the dynamometer and then the brake is applied at a variety of energy intensity levels and speeds. During this testing the drums (and other brake components) are instrumented to collect temperatures, stresses and pressures. Under the test conditions, severe conditions can be simulated and the performance of the drums can be evaluated both physically and thermally. Detailed surface analysis of the drums after the testing can provide evidence of their durability under simulated "real world" conditions.

5. Conduct Test Track Evaluation

Subsequent to a successful laboratory testing program, the drums will be evaluated on Canadian and US federally regulated braking tests (CMVSS/FMVSS 121) on a test track. This test involves the selection of a test vehicle (single unit truck) and the burnishing of the drums and pad material, followed by a series of prescribed brake events. Successful completion of this test indicates compliance of the brake system to Canadian and US federal vehicle safety regulations.

Following the successful completion of the CMVSS/FMVSS 121 test, it is recommended that a series of other performance and severe duty tests be undertaken, including Alpine descent, continuous drag, multiple high speed stops, etc. After each series of tests the drums will be physically evaluated for wear, thermal stress, etc. Subsequently, a brake structural integrity test should be done. This test exercises the brake system through 2000 stops based on SAE Test Method 1250. It provides a good severe test of the system's durability.

Finally, a series of tests should be undertaken using a severe braking test with progressive increases in brake imbalance. The goal of this stage of the test program is ultimately to cause a failure of the brake system and thereby understand the failure modes and effects of the materials and brake system.

6. Complete Evaluation

Throughout the brake testing program, physical and material inspection of the drums and friction material must be done to document evidence of heat or wear-related damage. Also, thermal history profiles can be collected to assess the improved thermal conditions offered by the MMC materials. A comparison to standard drum materials can be made to assess and quantify the physical and thermal performance benefits.

Composite NRCan Material

The evaluation of the NRCan composite material contains a number of additional steps. NRCan must undertake the work because of the proprietary nature of its material and processes. The work steps to create prototype drums are outlined in the following text. After the delivery of the prototype drums, the test procedures from MMC Task 3 on would be followed.

1. Fabricate Composite Test Samples

In this task, test samples made with CANMET hybrid composite and Duralcan material will be fabricated. Specifically, test samples suitable for the evaluation of wear behaviour, tensile strength, thermal conductivity, specific heat, elastic modulus, ductility and high temperature property will be produced.

2. Evaluate Composite Test Samples

Test samples produced in Task 1 will be subjected to property evaluation. Among the properties that will be evaluated are wear behaviour, tensile strength, thermal conductivity, specific heat, elastic modulus, ductility and high temperature property.

3. Select Drum Brake Rotor Type and Fabrication Process

In this task, the type of brake drum to be fabricated with composite materials will be selected. In addition, the fabrication processes used for making CANMET hybrid and Duralcan composite brake drums will be determined.

4. Fabricate Squeeze Cast Die

For the fabrication of brake drums using CANMET hybrid composite material, the squeeze casting technique must be employed. In this case, a steel squeeze casting die will need to be fabricated. Die attachments such as a heater, power supply, thermocouples, etc. will need to be purchased.

5. Fabricate Squeeze Cast Composite Drum Brake Rotors

Once the fabrication facility is set up, composite drum brake rotors, using both CANMET Hybrid composite and Duralcan materials, will be produced. It is intended that six composite brake drums (from each material) will be produced.

6. Analyse Data and Report Results

All information acquired in Tasks 1 to 5 will be documented and presented in a final report. The report will contain the following:

- results on the properties of CANMET hybrid and Duralcan composites;
- information on the manufacturing processes and procedures for composite brake drums; and
- technical constraints/challenges of the manufacturing processes for composite brake drums.

Test Program Costs and Schedule Estimates

The MMC-based prototype casting and testing program is estimated to cost C$100,000 (see cost breakdown in Table 5-2). Roughly half of this total budget is the direct costs associated with casting, finishing, testing and material analysis of the drums. The balance of the budget covers the costs of project management and travel associated with co-ordinating, monitoring and reporting on the test program.

The estimated project schedule is summarized in the Gantt chart in Figure 5-1. Time required for the castings is quoted as three weeks and the CMVSS/FMVSS 121 tests are normally completed in a two-week period. The extended track testing is estimated to require an additional four weeks. The total testing program is estimated to require six months to complete.

The NRCan product development program to develop and deliver six composite and six MMC prototypes is estimated at C$120,000 (see Table 5-3 for task costs) and will require approximately 12 months to complete. After prototype delivery, there will be an additional C$88,000 (and five months) required for the lab and test track evaluation program (C$100,000 minus the direct costs of the MMC casting C$12,000) for a total of C$208,000.

Table 5-2 MMC Drum Testing Program Cost Breakdown

Task	Title	Costs by Task (C$) Prj Mgr	Sub-Total Labour	Expense Travel	Misc*	Sub-Total	Project Total
	Per Diem Fees (8 h/day)	$1,120					
1	Finalize Testing Plan, Participants and Suppliers	$6,720	$6,720	$4,480		$4,480	$11,200
2	Cast Drums (12)	$0	$0	$0	$10,800	$10,800	$10,800
3	Machine and Finish Drums	$0	$0	$0	$1,500	$1,500	$1,500
4	Select Friction Material	$4,480	$4,480	$2,240	$2,000	$4,240	$8,720
5	Conduct Laboratory Brake Testing	$0	$0	$0	$7,200	$7,200	$7,200
6	Analyse Material	$3,360	$3,360	$0	$2,000	$2,000	$5,360
7	Test Track – CMVSS/FMVSS 121	$5,600	$5,600	$2,600	$12,000	$14,600	$20,200
8	Test Track - Extended Tests	$3,360	$3,360	$2,360	$15,000	$17,360	$20,720
9	Evaluate Material and Performance	$3,360	$3,360	$0	$3,000	$3,000	$6,360
10	Produce Report	$4,480	$4,480	$0	$500	$500	$4,980
	Total	**$31,360**	**$31,360**	**$11,680**	**$54,000**	**$65,680**	**$97,040**

* Misc. includes communications, courier, printing and other directly incurred costs.

10%	Contingency	$3,136
	Total	$100,176

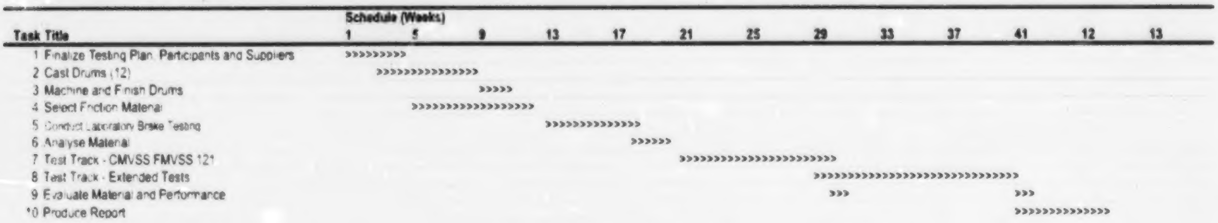

Figure 5-1 Test Program Schedule

Table 5-3 Cost Breakdown for NRCan Composite Brake Drum Fabrication

Major Task	Sub-Total (C$)
Fabricate Composite Test Samples	$10,000
Evaluate Composite Test Samples	$15,000
Select Drum Brake Rotor Type and Fabrication Process	$2,000
Fabricate Squeeze Cast Die	$30,000
Fabricate Squeeze Cast Composite Drum Brake Rotors	$45,500
Analyse Data and Report Results	$5,000
Materials & Supplies	$8,000
Travel	$3500
Total	**$120,000**

5.2.2 Product Development Partner Options

In order for a successful product to be developed and launched into the heavy-duty brake market, a number of key actors must be associated with the development project. These include:

Product Manufacturer/Distributor

As noted in the market analysis, the brake drum industry consists of a handful of OEMs that either supply just the drums or integrate the drum into a complete brake system, which is sold to the vehicle OEM or to the aftermarket. One of the OEM's primary roles is to accept the design responsibility and liability of the product. This is a substantial obligation and makes entry into the market by a start-up company difficult, if not impossible. Also, the OEM has developed and controls a market distribution chain of distributors, jobbers, etc. Again, this parts marketing and distribution system take a long time and substantial financial investment to create, making it essential for any new product to be developed in conjunction with a major OEM such that the new product can enter the market with as much technical credibility and market impetus as possible.

During the study, the project team met or interviewed most of the current OEMs supplying the North American market. The most recent experience with MMC was found to be at Hayes Lemmerz which, while having "shelved" its product development, was open to collaboration on a revised design and material for its MMC brake drum. At present, the company is open to consideration of an evaluation project and would consider contributing its existing mould patterns, laboratory test time and technical support to the project.

Casting Material Supplier

There are a number of choices for the supplier of MMC as outlined in the production cost analysis. At the proof of concept stage it is relatively unimportant as to which of the North American-based suppliers' material is used (Duralcan or MC-21). Based on Volvo's experience, it appears physically possible and possibly better to use a SiC particle size mixture that is broader than that normally used in the Duralcan product. The advantage of the broader size spectrum is a reduced product cost (of no importance at the evaluation stage) and possibly increased matrix strength and thermal transfer properties. The downside of the larger particles is that proper mixing of the particles and keeping them in suspension during the casting process is more problematic.

Preliminary discussions with both Duralcan and MC-21 have proved positive with both expressing interest in the evaluation program.

Caster

The choice of caster for the MMC material should be based on prior familiarity with MMC casting and heavy-duty drum designs. The previous casting of the Hayes Lemmerz drum was done by Eck Foundry and, for convenience and continuity, any new casting should be done with Eck. Other Ontario and Quebec aluminum casting plants have the capabilities to produce pilot or full production quantities when that stage is attained. Sourcing for the production casting is not considered a priority in this evaluation stage of the product development.

Friction Material Supplier

The other side of the friction couple is a key factor in the success of the MMC drum. The formulation of the pad material for good friction layer development, friction coefficients and wear is a difficult task. A number of friction material suppliers have been involved in MMC brake programs in the past and all are likely to be interested in participating in any heavy-duty program. None of the friction material suppliers are located in Canada.

Vehicle OEMs

The merits of lighter brake systems are well understood by the vehicle OEMs and a number of the larger truck manufacturers expressed interest in the potential product. The most energetic response was obtained from Oshkosh Trucks (Wisconsin), which stated that it wished to participate in any trial of the brake drums. Oshkosh saw an immediate application in its cement mixer product lines and possibly in its military vehicles. No other OEM (bus or truck) was this positive in its consideration of the product and evaluation project.

The selection of the initial test vehicle is somewhat unimportant except that it should represent a Class 8 tandem axle truck. The ability to use the vehicle to simulate an urban bus application would be beneficial and could be achieved through either disablement of the second rear brake set or the availability of a bus test vehicle.

End User

For the prototype test and evaluation stage, it is too early to heavily involve end users. However, the development of a user group that could provide test and brake performance requirements advice to the program would be useful. The existence of this advisory group would also accelerate the development of a field demonstration, assuming that the product is found to be satisfactory in the controlled tests. The user group should include staff from severe duty application fleets (cement mixers, dump trucks, etc.) and transit properties.

Canadian Content Issues

The brake manufacturing industry is controlled by a handful of multinational companies, none of which are based in Canada. While there are casting companies located in Canada that provide brake components, most of these companies supply to the light-duty market. There are, at present, no known suppliers for heavy-duty brake drums (or discs) in Canada. There are a number of aluminum casters that have the capabilities to cast MMC (or the NRCan composite material) and these companies could be developed into brake part suppliers if the product is commercialized.

For the prototype testing phase, most of the resources and partners are non-Canadian. Parts of the work, such as project management and track testing, could be Canadian sourced. The value of these tasks will be in excess of 50 percent of the total project value. Also, it is probable that commitments to Canadian sourcing of casting or materials could be negotiated based on provision of Canadian government financial investments in the further product testing and development phase.

5.3 Recommendations

Based on the analysis undertaken in this study, it has been concluded that the use of MMC materials in heavy-duty vehicle brake drums should be both physically feasible and financially attractive. However, the physical performance has not yet been demonstrated and thus it is recommended that a prototype design, manufacture and test program be undertaken in an attempt to demonstrate the physical performance of the material.

The two candidate material forms (MMC and NRCan composite) require different investments of resources to complete preliminary performance tests of the prototype designs. The analysis of the prototype testing costs and time schedules indicates that the MMC-based product would be less expensive and faster than the NRCan composite material, and have a lower development risk.

It is recommended that:

- A program should be developed in conjunction with a brake drum manufacturer to undertake the MMC-based drum test and evaluation program outlined above. Based on early discussions, substantial funding support could be provided by the brake drum OEM and friction material suppliers.
- Because the NRCan design has significant technical and economic merit, and if additional funding were available, it should also be supported as it potentially provides a less expensive manufacturing process and could result in higher Canadian content because of the intellectual property rights that NRCan has in the process.
- Based on the successful completion of a test of the prototype material(s), a larger demonstration of the product should be developed and more detailed production planning and manufacturing cost analysis be undertaken.

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